فيزياء المباني

أ.د/هشام جريشة

Building Physics

Heattransfer and Acoustics in Buildings

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To the People who I get from them my spirit

Chapter 1

Heattransfer

Preface

Buildings do not exist in isolation; they interact with the site on which they are built. Sustainable buildings provide high-quality, energy efficient, healthy and productive environments through the design and operation of innovative, high-performance process. Envelopes are implemented to achieve high-performance barriers, its components and systems that manage thermal loads and facilitate daylighting; Quantify the performance of innovative building envelope components and systems in terms of both energy and occupant impacts.

This book deals with thermal, acoustic and buildings energy saving. That means topics of building physics. This book intended for use as a textbook by undergraduate engineering students; and as reference book for practicing engineers.

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I would like to aknowledge with appreciation people like Professor K.Gertis, professor of building physics-Frauenhofer Institut, Germany, Professor K.Hauck, Professor of Building physics- Stuttgart University for the numerous and valuble teaching process.

لاتوجد المباني بمعزل عن محيطها الخارجي. ولا نجاح لتصميم أي مبنى إذا لم يهتم بقضايا العزل الحراري والعزل الصوتي ، فضلا عن كيفية الفراغ الداخلي من حيث الصحة والطاقة والإرتياح.
كل هذه العناصر تمثل تصميم عازل خارجي وهو الجدار ، والذي يعتبر بالنسبة لمحتويات المبنى الداخلية ليس إلا غلافا خارجيا. وتندرج كل هذه القضايا تحت مسمى **فيزياء المباني**.

ولا يسعني في هذا المقام سوى أن أتقدم بالشكر والعرفان لمن كانوا سببا في حبي لهذا العلم عبر الإستماع لمحاضراتهم الدسمة في رحاب جامعة شتوتجارت فترة الثمانينات من القرن الماضي.

أناس أسهموا بشكل فعال في تكويني العلمي ، على رأسهم الأستاذ الدكتور كارل جرتس والأستاذ الدكتور كارل هوك . كما أتقدم بالشكر أيضا لكل من ساهم في إخراج هذا الكتاب في نسخته تلك وعلى رأسهم الناشر مكتبة الأنجلو المصرية.

ربي اجعل هذا الكتاب في ميزان حسناتي
يوم لا ينفع مال ولا بنون إلا من أتى الله بقلب سليم

المؤلف
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Introduction

Providing thermal comfort without excess space conditioning cost is one of the primary requirements of buildings. Therefore, thermal control is an important aspect in almost all buildings. Understanding heat transfer and temperature distribution through building materials and assemblies is also important for assessing energy use, thermal comfort, thermal movements, durability, and potential moisture problems. Heat flow occurs through the building enclosure via opaque enclosure elements, is directly transferred into the building by solar radiation through windows, is carried along with air across the enclosure by unintentional leakage and ventilation, and can be generated within the building by occupants and their activities. The control of heat flow in buildings requires insulation layers compromised with few thermal bridges, an effective air barrier system, good control of solar radiation, and management of interior heat generation.

Until about 1850, the fields of thermos-dynamics and mechanics were considered two distinct branches of science, and the law of conservation of energy seemed to describe only certain kinds of mechanical systems. However, mid-19th century experiments performed by the Englishman James Joule and others showed that energy may be added to (or removed from) a system either by heat or by doing work on the system (or having the system do work). Today we know that internal energy, can be transformed to mechanical energy. Once the concept of energy was a universal law of nature

كانت ولاتزال الراحة الحرارية هدفا تصميما في العديد من المباني. ولهذا السبب تعد هي وغيرها من المصطلحات العلمية كجميع عمليات

النقـل الحـراري مـن توزيـع لـدرجات الحـرارة والإلمـام والدرايـة بمشــاكل الرطوبة من النقاط التي يعتمد عليها المصمم الحديث.

والحرارة تصل إلى داخل المباني عبر الإشعاع الشمسـي والتهوية وتزيد بزيادة ممارسـة المتواجـدين داخـل الفـراغ لنشــاط مـا. وبـذا يكـون لـدينا مصدرين أساسـيين للحرارة . والتحكم في الحرارة لا يتم إلا بوجود طبقة عازلة من الهواء أو الصوف الزجاجي.

وسـمك تلك الطبقة يتحدد على أسـاسـه درجـة حـرارة الفـراغ الـداخلي. أي أن المهندس يستطيع بالتحكم في سـمك الطبقة العازلـة أن يـتحكم فــي كميــة الحــرارة النافــذة إلــى الفــراغ الــداخلي أو المتســربة منــه. وبالتالي درجة الحرارة أيضا.

What is heat?

In physics and thermodynamics, heat is the process of energy transfer from one body or system to another due to a difference in temperature.[1] In thermos-dynamics, the quantity TdS is used as a representative measure of the (inexact) heat differential δQ, which is the absolute temperature of an object multiplied by the differential quantity of a systems entropy measured at the boundary of the object.

A related term is thermal energy, loosely defined as the energy of a body that increases with its temperature. Heat is also loosely referred to as thermal energy, although many definitions require this thermal energy to actually be in the process of movement between one body and another to be technically called heat (otherwise, many sources prefer to continue to refer to the static quantity as thermal energy). Heat is also known as Energy.

Heat flow can be a transient or a steady process. In the transient state, temperature and/or heat flow vary with time. Steady-state

heat flow occurs when the temperature and heat flow reach a stable equilibrium condition that does not vary with time.

Depending on the particular problem, the assumption of steady-state conditions may provide sufficiently accurate predictions of actual heat flow and temperature conditions may provide sufficiently accurate predictions of actual heat flow and temperature conditions. However, for some problems the assumption of steady flow can result in significant errors.

Heat flow can occur in one, two or three dimensions. In almost all real situations, heat flow occurs in three dimensions but, from a practical point of view, it is often acceptable to simplify considerations to only one-dimensional, or series, heat flow.

Heat transfer occurs by three primary mechanisms, acting alone in some combination: conduction, convection, and radiation.

Changes in moisture state, although not strictly an energy transfer mechanism must also be considered since these state changes absorb and release heat energy, i.e., latent heat.

في علم الديناميكا الحرارية تعرف الحرارة على أنها عملية إنتقال الطاقة أو صورة من صور الطاقة. وعملية إنتقال الحرارة يمكن لها أن تظهر في صورة ثابتة أو صورة متحركة.

في الصورة الثابتة تبدأ عمليات النقل الحراري عندما تصل إلى حالة الإتزان الثابت التي لاتتغير بتغير الزمن. والنقل الحراري يمكن له أن يكون أحادي أو ثنائي أوثلاثي الأبعاد.

Heat and internal energy

Internal energy is all the energy of a system that is associated with its microscopic components – atoms and molecules- when viewed from a reference frame at rest with respect to the object. The last part of

this sentence ensures that any bulk kinetic energy of the system due to its motion through space is not included in internal energy.

Internal energy includes kinetic energy of translation, rotation, and vibration of molecules, potential energy within molecules, and potential energy between molecules. It is useful to relate internal energy to the temperature of an object, but this relationship is limited. The internal energy changes can also occur in the absence of temperature changes. The internal energy of a monatomic ideal gas is associated with the translational motion of its atoms. This is the only type of energy available for the microscopic components of this system. In this special case, the internal energy is simply the total kinetic energy of the atoms of the gas; the higher temperature of the gas, the greater the average kinetic energy of the atoms and the greater the internal energy of the gas. More generally, in solids liquids, and molecular gases, internal energy includes other forms of molecular energy. For example, a diatomic molecule can have rotational kinetic energy, as well as vibrational kinetic and potential energy.

Heat is defined as the transfer of energy across the boundary of a system due to a temperature difference between the system and its surroundings.

When you heat a substance, you are transforming into it by placing it in contact with surroundings that have a higher temperature. This is the case, for example, when you place a pan of cold water on a stove burner – the burner is at a higher temperature than the water and so the water gains energy. We shall also use the term heat to represent the amount of energy transferred by this method.

Scientists used to think of heat as a fluid called caloric, which they believed was transferred between objects; thus, they defined heat in terms of the temperature changes produced in an object during heating. Today we recognize the distinct difference between internal energy and heat. Nevertheless, we refer to quantities using the names that do not quite correctly define the quantities but which have become entrenched in physics tradition based on these early ideas. Examples of such quantities are latent heat and heat capacity.

والطاقة الداخلية لنظام ما هي الموجـودة فـي مجمـوع جزيئاتـه ، الـذرة والجزيئـي . والطاقـة الكامنـة وطاقـة الوضـع همـا المسـببان للحركـة والإهتزاز داخل جزيئات الذرة . وبالمناسـبة ليسـت الحـرارة الصـادرة عـن الشـمس والتـي هـي ليسـت إلا تصـادم ذرات الهيـدروجين إلا ترجمـة حقيقة تظهر أن الحرارة ليست إلا صورة من صور الطاقة .

وفي الحقيقـة ليسـت الحـرارة دائمـا مظهـرا للطاقـة الكامنـة ، فالـذرات الأحادية لا تظهر طاقتها الكامنة بالتغير الحراري ولكن بإنتقال الإليكتـرون من مستوى طاقة إلى مستوى آخر.

As an analogy to the distinction between heat and internal energy, consider the distinction between work and mechanical energy not discussed in this book. The work done on a system is a measure of the amount of energy transferred to the system from its surroundings, whereas the mechanical energy of the system (kinetic or potential, or both) is a consequence of the motion and relative positions of the members of the system. Thus, when a person does work on a system, energy is transferred from the person to the system. It makes no sense to talk about the work of a system- one can refer only to the work done on or by a system when some process has occurred in which energy has been transferred to or from

the system. Likewise, it make no sense to talk about the heat of a system- on can refer to heat only when energy has been transferred as a result of a temperature difference. Both heat and work are ways of changing the energy of a system.

It is also important to recognize that the internal energy of a system can be changed even when no energy is transferred by heat. For example, when a gas is compressed by a piston, the gas is warmed and its internal energy increases, but no transfer of energy by heat from the surroundings to the gas has occurred. If the gas then expands rapidly, it cools and its internal energy decreases, but no transfer of energy by heat from it to the surroundings has taken place. The temperature changes in the gas are due not to a difference in temperature between the gas and its surroundings but rather to the compression and the expansion. In each case, energy is transferred to or from the gas by work, and the energy change within the system is an increase or decrease of internal energy. The changes in internal energy in these examples are evidenced by corresponding changes in the temperature of the gas.

ويمكـن لنـا أن نشـاهد أن الغـاز المضـغوط بفعـل المكـبس قـد ارتفعـت حرارته وتولدت طاقته الداخلية لكنها لاتنتقل . وإذا مـا تمـدد الغـاز بفعـل البرودة لا تنتقل الحرارة أيضا إلـى المحـيط الخـارجي . وبـذا يكـون زيـادة وانخفاض الطاقة الداخلية في هذا المثال داخل النظام.

Units of Heat

As we have mentioned early studies of heat focused on the resultant increase in temperature of a substance, which was often water. The early notions of heat based on caloric suggested that the flow of this fluid from one body to another cause changes in temperature. From the name of this mythical fluid, we have an energy unit related no

thermal processes, the calorie (cal), which is defined as the amount of energy transfer necessary to raise the temperature of 1g of water from14,5°c to 15,5°c .

The unit of energy in the British system is the British thermal unit (Btu), which is defined as the amount of energy transfer required to raise the temperature of 1 liter water from 63°F to 64°F.

بشكل دقيق يوجد وحدتان للقياس ، وحدة ال cal وهي الأكثر تداولا ، ووحدة ال Btu وهي الأقل تداولا . وتعريف كل واحة منهم هي القدرة على رفع كم معين من المياة بمقدار معين من درجة الحرارة .

Heat capacity and specific heat

When energy is added to a substance and no work is done, the temperature of the substance usually rises. An exception to this statement is the case in which a substance undergoes a change of state -also called a phase transition- the quantity of energy required to raise the temperature of a given mass of a substance by some amount varies from one substance to another.

The specific heat c of a substance is the heat capacity per unit mass. Thus, if energy Q transferred by heat to mass m of a substance changes the temperature of the sample by Δt, then the specific heat of the substance is

$$C = \frac{Q}{m\,\Delta T}$$

Specific heat is essentially a measure of how thermally insensitive a substance is to the addition of energy. The greater a materials

specific heat, the more energy must be added to a given mass of the material to cause a particular temperature change.

Substance	Specific heat at 25°c in J/kg°c	Specific heat at 25°c in cal/g°c
Aluminum	900	0.215
Cadmium	230	0.055
Copper	387	0.092
Silicon	703	0.168
Silver	234	0.056

Glass	837	0.200
Marble	860	0.21
Wood	1700	0.41

Alcohol	2400	0.38
Mercury	140	0.033
Water	4186	1.00
Steam	2010	0.48

For example, the energy required to raise the temperature of 0.500kg of water by 3°c is

$$\mathbf{0.5\ kg \times 4186\ J/kg°c \times 3°c = 6.28 \times 10^{3}\ J}$$

Note that when the temperature increases, Q and Δt are taken to be positive, and energy flow into the system. When the temperature decreases Q and Δt are negative, and energy flows out of the system.

Specific heat is varies with temperature. However, if temperature intervals are not too great. The temperature variation can be ignored

and c can be treated as a constant. For example, the specific heat of water varies by only about 1% from 0°c to 100°c at atmospheric pressure. Unless stated otherwise, we shall neglect such variations.

ليســت الحــرارة النوعيــة إلا عبــارة عــن الســعة الحراريــة لعنصــر مــا إذا اكتســب حــرارة خارجيــة وفــي حالــة عــدم وجــود حركــة. وهــي تختلــف بإختلاف العنصر.

Work and heat in thermodynamic processes

In the macroscopic approach to thermodynamics, we describe the state of a system using such variables as a pressure, volume, temperature, and Internal energy. The number of macroscopic variables needed to characterize a system depends on the nature of the system. For a homogeneous system such as a gas containing only one type of molecule, usually only two variables are needed. However, it is important to note that a macroscopic state of an isolated system can be specified only if the system in thermal equilibrium internally.

In the case of a gas in a container, internal thermal equilibrium requires that every part of the gas be at the same pressure and temperature.

Consider a gas contained in a cylinder fitted with a movable piston. At equilibrium, the gas occupies a volume V and exerts a uniform pressure P on the cylinders walls and on the piston. If the piston has a cross-sectional area A, the force exerted by the gas on the piston is

F = P x A

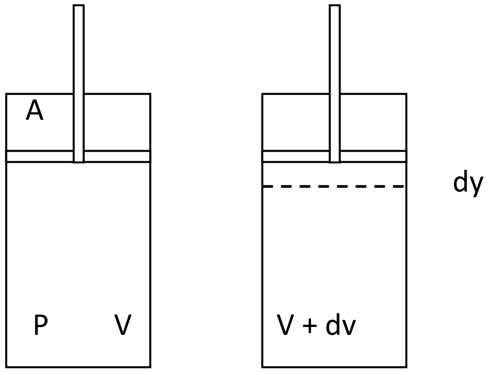


Fig: 1.1 the relation between the area of the piston and the pressure of the gas

Now let us assume that the gas expands quassi-statically, that is, slowly enough to allow the system to remain essentially in thermal equilibrium at all times as the piston move up a distance dy, the work done by the gas on the piston is :

$$\mathbf{dw = F . dy = P . A . dy}$$

because A.dy is the increase in volume of the gas dv, we can express the work done by the gas as:

$$\mathbf{dw = P . dv}$$

because the gas expands dv is positive, and so the work done by the gas is positive. If the gas were compressed, dv would be negative, indicating that the work done by the gas (which can be interpreted as work done on the gas) was negative.

إن الشغل الناتج عن حركة المكبس ليس إلا القوة مضروبه في فارق المسافة أو الضغط في مسطح المكبس في فارق المسافة. وإذا كان الشغل إيجابيا كانت الطاقة المبذولة خارج النظام ، وإذا كان سلبيا كانت الطاقة داخلية.

In the thermodynamics problems that we shall solve, we shall identify the system of interest as a substance that exchanging energy with the environment.

In many problems, this will be a gas contained a vessel; however, we will also consider problems involving liquids and solids. It is an unfortunate fact that, because of the separate historical development of thermo-dynamics and mechanics, positive work for a thermodynamic system is commonly defined as the work done by the system, rather than that done on the system. This is the reverse of the case for our study of work in mechanics. Thurs, **in thermodynamics, positive work represents a transfer of energy out of the system.** We will use this convention to be consistent with common treatments of thermodynamics.

The total work done by the gas as its volume changes from Vi to V1 is given by the integral of equation.

$$W= \int_{vi}^{v1} p.\, dv$$

To evaluate this integral, it is not enough that we know only the initial and final values of the pressure. We must also know the pressure at every instant during the expansion; we would know this if we had a functional dependence of P with respect to V.

ونستطيع أن نعبر عن الشغل بمعادلة تفاضلية فحواها أن الشغل ليس إلا الضغط مع تكامل الحجم.

This important point is true for any process the expansion we are discussing here, or any other. To fully specify a process, we must know the values of the thermodynamic variables at every state through which the system passes between the initial and final states.

In the expansion we are considering here, we can plot the pressure and volume at each instant to create a PV diagram like the one shown in the commend figure

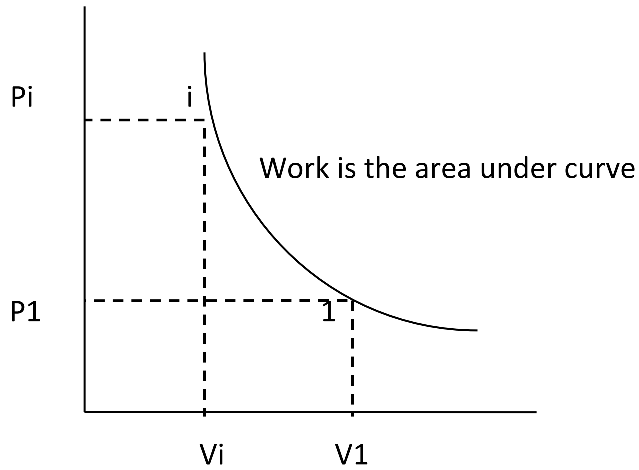


Fig: 1.2 the definition of the work according the thermodynamic process

As this figure shows, the work done in expansion from the initial state I to the final state 1 depends on the path taken between these two states, where the path on a PV diagram is a description of the thermodynamic process through which the system is taken. To illustrate this important point, consider several paths connecting I and 1 (fig). in the process depicted in the figure , the pressure of the gas is first reduced from Pi to P1 by cooling at constant volume vi. The gas then expands from Vi to V1 at constant pressure P1. The value of the work done along this path is equal to the area of the shaded rectangle, which is equal to P1 (v1-vi), the gas first expands from vi to v1 at constant pressure pi then, its pressure is reduced to pi at constant volume vi. The value of the work done along this path

is pi(v1-vi), which is greater than that for the process described in the fig1.3

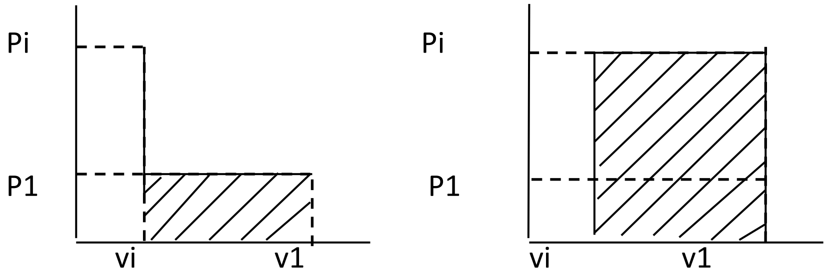


Fig:1.3 The value of the work along the path

Therefore, we see that the work done by a system depends on the initial and final states and on the path followed by the system between these states.

The energy transfer by heat **Q** into or out of a system also depends on the process, consider the situations depicted. In each case, the gas has the same initial volume, temperature, and pressure and is assumed to be ideal. The gas is thermally insulated from its surroundings except at the bottom of the gas-filled region, where it is in thermal contact with an energy reservoir. An energy reservoir is a source of energy that is considered to be so great that a finite transfer of energy from the reservoir does not change its temperature. The piston is held at its initial position by an external agent a hand, for instance. When the force with which the piston is held is reduced slightly, the piston rises very slowly to its final position. Because the piston is moving upward, the gas is doing work on the piston. During this expansion to the final volume v1, just

enough energy is transferred by heat from the reservoir to the gas to maintain a constant temperature Ti

إن التغير الحراري الناشئ له نقطة بداية Vi ونقطة نهاية V1 ، وكذلك بالنسبة لغير الضغط داخل الغاز الموجود تحت المكبس . وينتج عن ذلك أيضا تغير في درجة الحرارة . والطاقة الداخلية هي تلك الناشئة بسبب هذه المتغيرات الثلاثة.

Now consider the completely thermally insulated system. When the membrane is broken, the gas expands rapidly into the vacuum unit it occupies a volume v1 and is at a pressure P1. In this case, the gas doses no work because there is no movable piston on which the gas applies a force. **Furthermore, no energy is transferred by heat through the insulating wall.**

The initial and final states of the ideal gas are identical to the gas does work on the piston, and energy is transferred slowly to the gas. In the second case, no energy is transferred, and the value of the work done is zero. **Therefore, we conclude that energy transfer by heat, like work done, depends on the initial, final and intermediate states of the system**. In other words, because heat and work depend on the path, neither quantity is determined solely by the end points of a thermodynamic process.

The first law of thermodynamics

The law of conservation of mechanical energy state that **the mechanical energy of a system is constant in the absence of non-conservative force such as a friction.** That is, we did not include changes in the internal energy of the system in this mechanical model. The first law of thermodynamics is a generalization of the law of conservation of energy that encompasses changes in internal

energy. It is a universally valid law that can be applied to many processes and provides a connection between the microscopic and macroscopic worlds.

إن القــانون الأول للــديناميكا الحراريــة لــيس إلا قــانون حفــظ الطاقــة ، فالطاقة لا تفنى ولكنها تغير صورتها من شكل إلى آخـر . سيتضـح ذلـك من خلال طيات هذا الكتاب . ويمكن تطبيق ذلـك علـى عـالمي المـاكرو والميكرو ، أكبـر الأشـياء وأصغرها ، حتـى عـالم الـذرة والطاقـة الناتجـة منها.

We have discussed two ways in which energy can be transferred between a system and its surroundings. One is work done by the system, which requires that there be a macroscopic displacement of the point of application of a force (or pressure).

The other is heat, which occurs through random collisions between the molecules of the system. Both mechanisms result in a change in the internal energy of the system and therefore usually result in measurable changes in the macroscopic variables of the system, such as the pressure, temperature, and volume of gas.

وقد سبق لنا إيضاح أن الطاقة تنقل عبر طريقين.

-الأول الشغل المبذول عبر الضغط وحركـة المكـبس والتغيـر فـي حجم الغاز .

-والثـاني عبـر الحـرارة المنتقلـة عبـر تغيـر فـي جزيئـات النظـام ومحيطه الخارجي .

To better understand these ideas on a quantitative basis, suppose that a system undergoes a change from the initial state to a final

state. During this change, energy transfer by heat Q to the system occurs, and work w is done by the system. As an example, suppose that the system is a gas in which the pressure and volume change from Pi and Vi to P1 and V1. If the quantity Q – W is measured for various paths connecting the initial and final equilibrium states, we find that it is the same for paths connecting the two states. We conclude that the quantity Q – W is deter-mined completely by the initial and final states of the system, and we call this quantity the change in the internal energy of the system. Although Q and W both depend on the path, the quantity Q-W is independent of the path. If we use the symbol Eint to represent the internal energy, then the change in internal energy ΔEint can be expressed as

$$\Delta E_{int} = Q - W$$

Where all quantities must have the same units of measure for energy. Equation $\Delta E=Q-W$

وإذا أردنا أن نكون أكثر دقة وإيضاحا فإننا نقول أن الطاقة في نظام ما ليست إلا طرح الحرارة من الشغل المبذول. وليس هذا إلا تعبير عن القانون الأول للديناميكا الحرارية.

Is known as the first-law equation and is a key concept in many applications. As a reminder, we use the convention that Q is positive when energy enters the system and negative when energy leaves the system, and that W is positive when the system does work on the surroundings and negative when work is done on the system.

When a system undergoes an infinitesimal change in state in which a small amount of energy dQ is transferred by heat and a small amount of work dW is done, the internal energy changes by a small amount

dE. Thus for infinitesimal processes we can express the first law equation as

$$dE = dQ - dW$$

The first-law equation is an energy conservation equation specifying that the only type of energy that changes in the system is the internal energy Eint. Let us look at some special cases in which this condition exists.

First, let us consider an isolated system- that is, one that dose not interacts with its surroundings. In this case, no energy transfer by heat takes place and the value of the work done by the system is zero; hence, the internal energy remains constant. That is , becauseQ = W =0, it follows that ΔE = 0 , and thus Eint,I = Eint1. We conclude that the internal energy Eint of an isolated system remains constant.

Next, we consider the case of a system (one not isolated from its surroundings) that is taken through a cyclic process that is, a process that starts and ends at the same state. In this case, the change in the internal energy must again be zero, and therefore the energy Q added to the system must equal the work W done by the system during the cycle. That is, in a cyclic process,

$$\Delta E_{int} = 0 \qquad \text{and} \qquad Q = W$$

إن النظام المعزول عن محيطه لا يوجد به إنتقال للحرارة وبالتـالي قيمـة الشغل المبذول صفر وبالتالي يكون فرق الطاقة يساوي صـفر. وينطبـق أيضا هذا الكلام على النظام الذي يبدأ وينتهي عند نفـس النقطـة، فـي هذه الحالة سيكون التغير في الطاقة الداخلية يساوي صفر.

Some applications of the first law of thermodynamics

Before we apply the first law of thermodynamics to specific systems, it is useful for us to first define some common thermodynamic

processes. An **adiabatic process** is one during which no energy enters or leaves the system by heat that is, Q=0. An adiabatic process can be achieved either by thermally insulating the system from its surroundings or by performing the process rapidly, so that there is little time for energy to transfer by heat. Applying the first law of thermodynamics to an adiabatic process, we see that

$$\Delta E_{int} = -W \text{ (adiabatic process)}$$

From this result we see that if a gas expands adiabatically such that W is positive, then ΔEint is negative and the temperature of the gas decreases. Conversely, the temperature of a gas increases when the gas is compressed adiabatically. Adiabatic processes are very important in engineering practice. Some common examples are the expansion of hot gases in an internal combustion engine, the liquefaction of gases in a cooling system, and the compression stroke in a diesel engine.

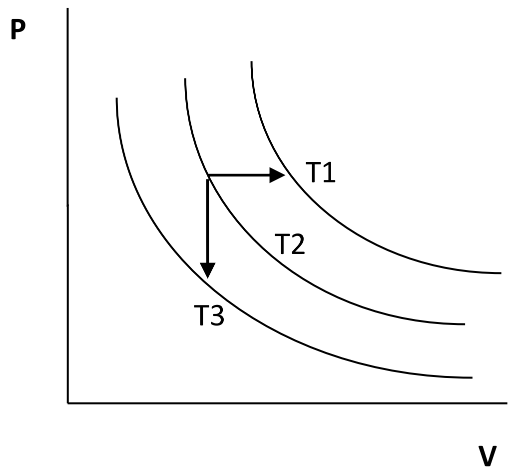


Fig:1.4 The adiabatic free expansion diagram

The process described in this diagram called an adiabatic free expansion, is unique. The process is adiabatic because it takes place in an insulated container.

نظام الأدياباتيك تنتقل فيه الطاقة كشغل دون إنتقال الحرارة . ويمكن التعبير عن ذلك بأن الطاقة تساوي سالب الشغل. بمعنى أنه إذا تمدد غاز بقيمة موجبة كشغل فإن الطاقة لابد أن تكون سالبة.

وتطبيقات الأديابتك كثيرة في اطار الديناميكا الحرارية منها وأهمها تمدد الغاز داخل ماكينة الإحتراق الداخلي.

Because the gas expands into a vacuum, it dose not apply a force on a piston as was depicted, so no work is done on or by the gas. Thus, in this adiabatic process, both Q=0 and W=0 . As a result, ΔEint=0 for this process, as we can see from the first law. That is, **the initial and final internal energies of a gas are equal in an adiabatic free expansion.** Aswe shall see, the internal energy of an ideal gas depends only on its temperature. Thus, we expect no change in temperature during an adiabatic free expansion. This predication is in accord with the results of experiments performed at low pressures.

Experiments performed at high pressures for real gases show a slight decrease or increase in temperature after the expansion. This change is due to intermolecular interactions, which represent a deviation from the model of an ideal gas.

A process that occurs at constant pressure is called an isobaric process. In such a process, the values of the heat and the work are both usually nonzero. The work done by the gas is simply

$$W = P\ (V1 - Vi) \quad \textbf{isobaric process}$$

Where P is the constant pressure.

A process that takes place at constant volume is called an **isovolumetric process**.In such a process, the value of the work done

is clearly zero because the volume dose not change. Hence, from the first law we see that in an isovolumetric process, because W=0

$$\Delta E_{int} = Q \quad \textbf{isovolumetric process}$$

ونستطيع القول أن الطاقة الناشئة والطاقة النهائية كطاقة كامنة لغاز من الغازات متساوية في حالة الأدياباتك الحر. ومن جهة أخرى إذا أضيفت طاقة بطريق الحرارة إلى نظام ما ترتفع الطاقة الناشئة في جميع أجزاء النظام

The expression specifies that if energy is added by heat to a system kept at constant volume, then all of the transferred energy remains in the system as an increase of internal energy of the system.

For example, when a can of spray paints is thrown into a fire, energy enters the system (the gas in the can) by heat through the metal walls of the can. Consequently, the temperature, and thus the pressure, in the can increase until the can possiblyexplodes.

A process that occurs at constant temperature is called an isothermal process. A plot of P versus V at constant temperature for an ideal gas yields a hyperbolic curve called an isotherm.

The internal energy of ideal gas is a function of temperature only. Hence, in an isothermal process involving an ideal gas, $\Delta E_{int}=0$. For an isothermal process, then we conclude from the first law that the energy transfer Q must be equal to the work done by the gas that is, Q=W. Any energy that enters the system by heat is transferred out of the system by work; as a result, no change of the internal energy of the system occurs.

Isothermal expansion of an ideal gass

Suppose that an ideal gas is allowed to expand quasi-statically at constant temperature, as described by the PV diagram. The curve is a hyperbola, and the equation of state of an ideal gas with T constant indicates that the equation of this curve is PV = constant.

The isothermal expansion of the gas can be achieved by placing the gas in thermal contact with an energy reservoir at the same temperature.

Let us calculate the work done by the gas in the expansion from state I to state 1 .(fig:1.2) The work done by the glass is given by equation

$$\mathbf{W = \ p.\ dv} \int_{vi}^{v1} p$$

Because the gas is ideal and the process is quassi-static, we can the expression PV=nRT for each point on the path.

Therefore, we have

$$\mathbf{W=} \int_{vi}^{v1} p \ \mathbf{.\ dv =} \int_{vi}^{v1} \frac{nRT}{v} \mathbf{.\ dv}$$

Because T is constant in this case, it can be removed from the integral along with n and R:

$$\mathbf{W = nRT} \int_{vi}^{v1} \frac{dv}{v} \mathbf{dv = nRT.\ LnV}$$

To evaluate the integral, we used $\int {dx}/{x}$ = lnX. Evaluating this at the initial and final volume we have

$$\mathbf{W = nRT.\ Ln\ (} {v1}/{vi} \mathbf{)}$$

Numerically, this work W equals the shaded area under the pv curve. Because the gas expands V1 > Vi, and the value for the work done by the gas is positive, as we expect. If the gas is compressed, then V1 < Vi , and the work done by the gas is negative.

وعلى هذا فإن تمدد الغاز المثالي لنظام الأيزوثرمال بفرضية أن هذا التمدد يتم عند درجة حرارة ثابتة يكون الشغل الناتج علاقة تكاملية بين الحجم والضغط. إن المتأمل في إنجازات القرن الثامن عشر يجد أنها تدور حول القانون الأول من علم الثرموديناميك من علاقة بين الحجم والضغط .تخيل أن تلك العلاقة حركت كتل من الحديد وجعلت سيارة بوزن 700 كيلو جرام تسير بسرعة مائة كيلو متر. يضاف إلى ذلك تحولات الطاقة من كيميائية إلى حركية في صفر ثانية.

Example A

1.0 mol. Sample of an ideal gas is kept at 0.0°c during an expansion from 3.0L to 10.0L . How much work is done by the gas during the expansion.

Solution

$$W = nRT.\ \mathrm{Ln}\left(\frac{V1}{Vi}\right)$$

$$W = (1.0\mathrm{mol})(8.31\mathrm{J/mol.k})(273\mathrm{K})\ .\ \ln\left(\frac{10}{3}\right)$$

$$= 2.7 \times 10^{3}\ \ \mathrm{J}$$

(b) How much energy transfer by heat occurs with the surrounding in this process?

From the first law we find that

$\Delta Eint = Q - w$

$0 = Q - W$

$Q = W = 2{,}7 \cdot 10^3$ J

(c) If the gas is returned to the original volume by means of isobaric process, how much work is done by the gas?

The work done in an isobaric process is given by the equation. We are not given the pressure, so we need to incorporate the ideal gas law

$$\mathbf{W = P(V1\text{-}Vi) = \frac{nRT}{Vi}(V1\text{-}Vi)}$$

$$= \frac{(1.0 mol)\left(\frac{8.31 J}{mol}.k\right)(273 k)}{10x\ 10^{-3} m^3}$$

$= -\ 1.6 \times 10^3$ J

We use the initial temperature and volume to determine the value of the constant pressure because we do not know the final temperature. The work done by the gas is negative because the gas is being compressed.

Example B

Suppose 1.00g of water vaporizes isobarically at atmospheric pressure 1.013×10^5 pa . its volume in the liquid state is Vi=Liquids

Vi=Liquids =1.00 cm^3, and its volume in the vapor state is V1=V vapor = 1.671 cm^3. find the work done in the expansion and the change in internal energy of the system. Ignore any mixing of the steam and the surrounding air-imagine that the steam simply pushes the surrounding air out of the air.

Solution

Because the expansion takes place at constant pressure, the work done by the system in pushing away the surrounding air is, from the equation.

W = P(V1-Vi)

= (1.013 x 10^5pa)(1.67 x $10^{-6}m^3$ - 1.00

X $10^{-6}m^3$ = 169 J

To determine the change in internal energy, we must know the energy transfer Q needed to vaporize the water

Q = m.L = 1.00 x 10^{-3}kg) (2.26 x 10^6J/kg)=2260J

Hence, from the first law, the change in internal energy is

ΔE = Q –W = 2260 J – 169 J = 2.09 kJ

The positive value for ΔE indicates that the internal energy of the system increases. We see that most 2090J/2260J = 93% of the energy transferred to the liquid goes into increasing the internal energy of the system. Only 169J/ 2260J = 7% leaves the system by work done by the steam on the surrounding atmosphere.

Example C

A 1.0 kg bar of copper is heated at atmospheric pressure. If its temperature increases from 20°c to 50°c (a) what is the work done by the copper on the surrounding atmosphere.

Solution

Because the process is isobaric, we can find the work done by the copper using the equation W= P(v1-vi) . We can calculate the change in volume of the copper

$$\Delta V = \beta \,.\, V_i \,.\, \Delta T$$

$$= (5.1 \times 10^{-5})^{-1} (50°c-20°c)\, Vi = 1.5 \times 10^{-3} Vi$$

The volume Vi equal to m/ρ from the table that the density of copper is 8.92 x 10^3 kg/m^3. Hence,

$$\Delta V = (1.5 \times 10^{-3}) \left(\frac{1.0\ kg}{8.92 \times 10^3\ kg/m^3} \right) = 1.7 \times 10^{-7} m^3$$

The work done is

$$W = P \,.\, \Delta V$$

$$= (1.013 \times 10^5 N/m^2)(1.7 \times 10^{-7} m^3)$$

$$= 1.7 \times 10^{-2} J$$

(b) What quantity of energy is transferred to the copper by heat ?

Taking the specific heat of copper we find that the energy transferred by heat is

$$Q = m . c . \Delta T$$

$$= 1.0 \text{ kg} \times 387 \text{J/kg.°c} \times 30°c = 1.2 \times 10^4 \text{ J}$$

(c) What is the increase in internal energy of copper?

$$\Delta E = Q - W = 1.2 \times 10^4 - 1.7 \times 10^{-2} = 1.2 \times 10^{-2} \text{J}$$

Note that almost all of the energy transferred into the system by heat goes into increasing the internal energy. The fraction of energy used to do work on the surrounding atmosphere is only about 10^{-6}. Hence, when analyzing the thermal expansion of a solid or liquid the small amount of work done by the system is usually ignored.

إن الأمثلة السابقة لخصت العلاقة بين الشغل والحجم ودرجة الحرارة والضغط . وما الحركة الناتجة أو الشغل المبذول إلا إنعكاس للطاقة المتولدة داخليا . أما بالنسبة للإحتكاك بين الجزيئات عند تمدد الغاز أو السائل يكون مهملا بسبب قيمته المتدنية 10^{-6}.

Thermal Transmission

There are three methods by which heat can be transferred.These processes are known as:

- Conduction
- Convection
- Radiation

1. Conduction

Is the flow of through a material by direct molecular contact? This contact occurs with a material or through two materials in contact. It is the most important heat transport mode for solids; it is some times important for liquids, and it is occasionally important for gases.

When heat is transferred through a solid substance, the molecules are unable to move and start to vibrate. This vibration is passed to the next molecule by a chain reaction. In this way heat is transferred through a material and process of conduction takes place.

في هذا الكتاب سنركز على النقل الحراري داخل جدار الغرفة ، سواء كان جدار داخلي أو خارجي . وسنتطرق بعمق إلى عمليات النقل الحراري الثلاث الأساسية . النقل بالتوصيل والإنتشار والإشعاع.

وتلك بداية بسيطة لكنها السلمة الأولى في علم فيزيا البناء . لنبدأ بالتفريق بين النقل بالتوصيل والنقل بالإنتشار . النقل بالتوصيل يكون في الجمادات بين طبقة وطبقة دون ملامسة الهواء. وفي المباني يكون النقل بالتوصيل بين طبقات الجدار المختلفة دون ملامسة الهواء أيضا. أما إذا انتقلت الحرارة من الجدار إلى الهواء الداخلي أو الخارجي فإن ذلك يسمى النقل بالإنتشار.

If a piece of copper tube is heated at one end, the other end will sooner or later get warm. Heat has passed along the piece of copper by the process of conduction.

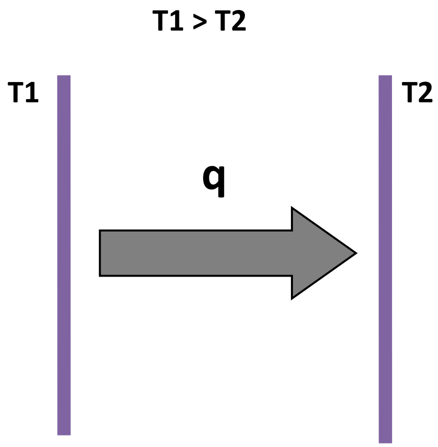


Fig: 1.5 Heattransmission from layer to another

Heat Conduction Pathways

This illustration shows the cross section of a building that many would consider to be energy-efficient: 2x6 studs, "house wrap" on the exterior, thick roof insulation, and insulated windows. As the arrows show, there can still be many places where substantial heat loss can occur through conduction.

Fourier's law of conduction

The law of heat conduction, also known as Fourier's law states that the time rate of heat transfer through a material is proportional to the negative gradient in the temperature and to the area at right angles, to that gradient, through which the heat is flowing. We can state this law in two equivalent forms: the integral form, in which we look at the amount of energy flowing into or out of a body as a whole, and the differential form, in which we look at the, flows or fluxes of energy locally.

Electrons in a metal transfer the heat from one particle to another further away. A simpler way to describe it is that it is heat transferred through touch and heat from friction.

قانون فورير

إن الوقـت المفقـود فـي إنتقـال حـرارة عبـر عنصـر يتناسـب مـع درجـات الحرارة السـالبة ومع المسـطح . والمقصـود بـدرجات الحـرارة السـالبة K ليس درجات حرارة ،لكنها معـاملات التوصـيل يضـاف إليهـا فـارق درجـات الحرارة.

Differential form

The differential form of Fourier's law of thermal conduction shows that the local heat flux, $\vec{q}$, is proportional to the thermal conductivity, k, time the negative local temperature gradient,-∇T. The heat flux is the amount of energy that flows through a particular surface per unit area per unit time

$$\vec{q} = -k. \nabla T$$

Where (including the SI units)

The thermal conductivity, k, is often treated as a constant, though this is not always true. While the thermal conductivity of a material generally varies with temperature, variation can be small over a significant range of temperatures for some common materials. In anisotropic materials, the thermal conductivity typically varies with orientation; in this case k is represented by a second-order tensor. In non uniform materials, k varies with spatial location.

For many simple applications, Fourier's law is used in its one - dimensional form. In the x-direction,

$$\mathbf{qx = -k} \frac{dt}{dx}$$

Integral form

By integrating the differential form over the materials total surface S, we arrive at the integral form of Fourier's law:

$$\frac{dq}{dt} = -k\oint_S \vec{\nabla} T.\vec{dS}$$

Where (including the SI units)

$\frac{dq}{dt}$ is the amount of heat transferred per unit time [w] and ds is an oriented surface area element [m2]

The above differential equation, when integrated for a homogeneous material of 1-Dgeometry between two endpoints at constant temperature, gives the heat flow rate as:

$$\frac{\Delta q}{\Delta t} = -k.A.\frac{\Delta T}{\Delta x}$$

Where:

A is the cross-sectional surface area,

ΔT is the temperature difference between the ends,

Δx is the distance between the ends.

This law forms the basis for the derivation of the heat equation. R-value is the unit for heat resistance, the reciprocal of the conductance. Ohm's law is the electrical analogue of Fourier's law.

Conductance

Many solids such as common brick, wood siding, thermal batt or board insulation, gypsum board, and so on are widely available in standard thicknesses. For such common materials, it is useful to know the rate of heat flow for that standard thickness instead of the rate per meter. Conductance, designated as c, is the number of Btu per hour that flow through 1 square meter a given thickness of material when the temperature difference is 1°c. The units are w/m^2K

Conductance can be written:

$$\mathbf{U} = \frac{k}{\Delta x}$$

Where U is the conductance in W/m^2K

Fourier's law can also be stated as:

$$\mathbf{Q = U.A.\Delta\delta}$$

Conductivity

Each material has a characteristic rate at which heat will flow through it. For such individual, or homogeneous, solids, this rate is called conductivity.

Material	Conductivity [W/mk}
Aluminium	238
Steel	52
Granit	2,9
Wood	0,14

When the temperature drop through that material is 1k (equal to 1°c) under conditions of steady heat flow. Conductivity is an important factor in passive heating or cooling design that depends heavily on the rate at which heat is conducted into a material from its surface. When we touch a solid's surface, we sense its conductivity (rather than its temperature). Conductivity for each solid is established by tests and is published as a basic rating.

$$K = [\frac{1}{\alpha i} + \sum si/\lambda i + \frac{1}{\alpha o}]^{-1}$$

لكل شريحة من شرائح الحائط معامل توصيل $\sum si/\lambda i$ يختلف بإختلاف المادة. وحاصل جمع الشرائح مقسومة على سمك الطبقة أو الشريحة ناتجه هو ال U-Value

Heat flow through air

At the exposed surfaces of solids, heat transfer takes place both by convection and radiation. Evaporation also can occur, occasionally with thermally significant results; see fig. 1.6 (British Pavilion, Seville)

for an example of an entire wall so used. Convection is highly dependent on air motion, so wind outdoor must be considered. Also because warm air rises and cold air falls, vertical surfaces that encourage this kind of airflow exchange heat faster than the same surfaces placed horizontally, unless the of the heat flow is upward through this horizontal layer.

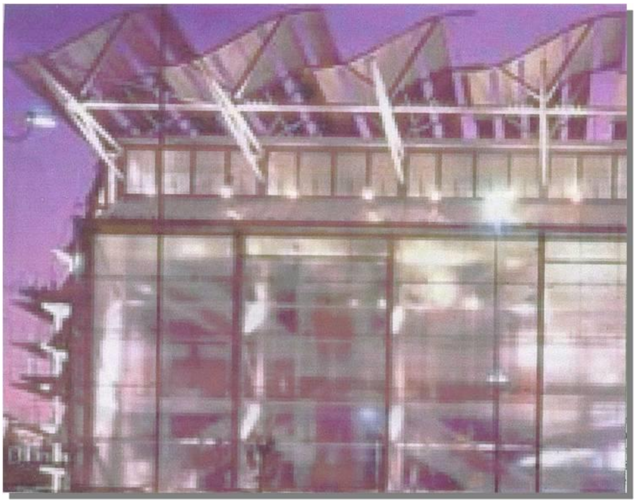


Fig:1.6 British Pavilion from Nicolas Grimshow

أولا أعتذر للقارئ الكريم ، طالبا كان أم مهندسا ، أنني لا أستخدم العربية في كل فقرة من فقرات الكتاب حتى لا يكبر حجمه ، وهو على كل الأحوال كبير. هذا فضلا عن أن لغة العلم وبلا شك هي اللغة الإنجليزية فأنا أجد صعوبة كبيرة في ترجمة العديد من المصطلحات العلمية دون أن تخرج ركيكة مبتذلة.

ثانيا هذا مشروع رائع للمهندس جريم شو عالج فيه الزجاج معالجة فيزيائية بشلال من المياة الهادئة أبان عن قدرة وبراعة في التصميم وقلل التسرب الحراري إلى داخل الفراغ في دولة مشمسة من دول حوض البحر الأبيض المتوسط –أسبانيا- . إن شو بهذ التصميم يعلم العديد من المهندسين كيف يكون التصميم يقف على قدم راسخة في

العلم. وقد قمت بتطوير هذا التصميم في رسالة الدكتوراة بجامعة شتوتجارت للحصول على درجة تبريد أعلى. راجع كتاب فقراء العمارة للمؤلف.

Resistance

When air motion along surfaces is minimal, an insulating layer of air is created. The resistance of this layer of still air along a vertical surface is numerically equal to that of a thickness of 12.7mm plywood. However when this air layer is disturbed, its resistance drops quickly; with 6,7m/s wind, resistance drops to about one quarter of the still-air value. Similar drops in resistance occur when forced-air grilles are located immediately above or below windows.

$$\boxed{R = \frac{1}{k} \quad [mk/w]}$$

This surface layer of air is often evaluated by their conductance, the reciprocal of resistance, because we often wish to encourage heat transfer between solids and air. For example, passive heating and cooling systems are dependent on large interior surfaces as heat exchangers. By contrast, active (conventional) systems typically concentrate on heat exchange within mechanical equipment.

Writing

$$U = \frac{k}{\Delta x}$$

Where U is the conductance, in $W/(m^2k)$.

Fourier's law can also be stated as:

$$Q = U.\ A.\ \Delta\delta$$

Insulation

The combination of dead (still) air space and reflective surfaces produces some of our most effective insulating products, especially when they are made of lightweight materials of low conductivity. Glass fiber, cellular glass expanded styrene's (foamed plastics), and mineral fiber all exhibit the characteristics of enclosing vast number of dead-air spaces per unit volume. When bonded to reflective films and properly installed (the shiny film facing a dead air space), high resistance to heat flow is achieved. Resistance to heat flow can be compared between various building materials, including insulation products and air films and air space.

Heat flow through the opaque building envelope

When the process of heat flow is understood, the calculation can begin. Initially, the hourly heat loss or heat gain is calculated, because these rates can later be used either to establish equipment size requirements (design conditions) or approximate energy consumption (average conditions). To calculate the hourly flow through a buildings envelope, these are necessary:

1. The rate at which heat flows through the various assemblies of materials that makes up the envelope.
2. The area of each of these assemblies.
3. The temperature difference between inside and outside for the hour being calculated.

Often, only the first two factors are calculated, which allows a comparison of one building with any other, anywhere. The third factor, temperature difference, is strongly climate related.

The areas of walls, roofs, windows, doors, and floors of various kinds are determined from preliminary architectural design drawings, final versions of these drawings should be subject to changes suggested by these calculations. Such changes are usually in roof, wall and other construction (affecting the rate of heat flow) rather than in area. The temperature differences for design conditions are listed in appendix A; for energy consumption.

U- Value

The variety of terms used so far to express heat flow is potentially bewildering these terms are, however, but part of a larger picture; what is needed is one final, overall expression of the steady-state rate which heat flows through architectural skin elements (wall, roofs, etc). This is provided by the U value (also called the U-factor). Where U is thermal transmittance, again expressed in the now-familiar terms of w/m^2k .

Because U values are both common and rather complex, several pages devoted to listing them for familiar constructions of walls, floors, roofs, doors and windows.

These U values are calculated for a particular element (roof,wall,etc) by finding the resistance of each of its component materials, including air layers and internal air spaces then adding all these

resistance to obtain ∑R. The U value is to reciprocal of this sun of resistance.

$$U = k = \sum \frac{1`}{R}$$

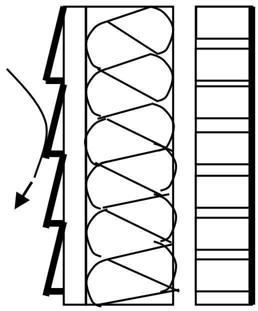


Fig: 1. 7 Procedures for determining the U- value for a wall with interior thermal mass.

<u>Pre-calculations</u>

inside air layer	0.68⅓-	in plywood	0.62
Common brick	0.20	1- in wood siding	0.79
Fiberglass 6 in	19.00	Outside air layer	0.17

$\sum R = 21.46$

$U = \frac{1}{\sum R} = \frac{1}{21.46}$ = 0,046 w/ m²k

The procedure is illustrated unfortunately; many common constructions are not so simple, as shown in the sections that follow.

"Pre-calculated" U values for many common constructions are found in tables

	Construction	Insulation R-value F ft h/Btu	Insulation R-value km²/w	Insulation U-value F ft h/Btu	Insulation U-value km²/w
Wall	Wood siding fiber board sheathing, blanket insulation, gypsum board	R-11	1.94	0.081	0.52
Roof	Built-up roof plywood deck, 2-in.and R-19 blanket insulation with no reflective air space, gypsum board, acoustic file	R-19	3.35	0.046	0.3
Floor	Metal lath and lightweight aggregate plaster, 2-in x 8-in. floor joist with R-19 blanket insulation and no reflective air space, wood subfloor, plywood, and floor file	R-19	3.35	0.052	0.3

Trends in U- and R values

Before looking individually at each component of the envelope, some history and prediction may be useful. As a result of oil embargo of the early 1970s, there have been huge increases in component R (therefore huge decreases in component U), sometimes because of increased thickness and insulation (as with walls and roofs), sometimes because of substantially changed materials (as with windows). What might the next quarter century bring, and how

might the designer of today's building envelope anticipate or participate these advances.

Structural systems are changing fundamentally due to advent of structural insulated panels. They promise greatly improved insulation and air-tightness , compared to the site-built

Walls

Compared to other surfaces of the building envelope, wall U values are quite straightforward. There are few complications such as ground contact or crawl spaces with floors, or intermediary attic spaces with roofs. There is, however, the matter of thermal bridging: where framing interrupts the insulation, an averaged U-value must be found.

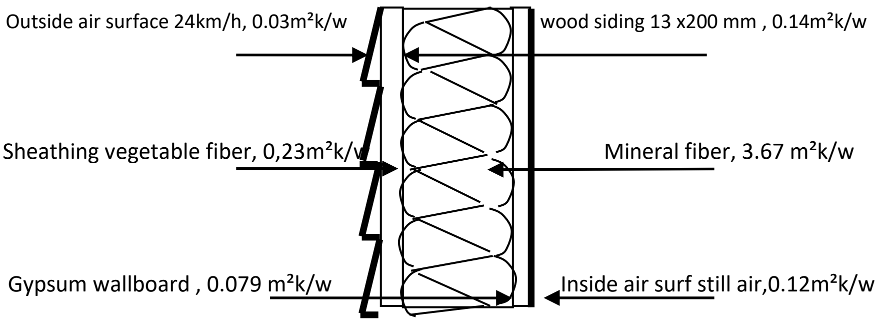


Fig: 1.8 R-Value for different Constructions

فيزيئيا ليست حدود المبنى الخارجية إلا غلاف أو ظـروف يحمـي الفـراغ الداخلي من عوامل التعريـة. وفـي التمـارين القادمـة سـنتعرض لحـالات مختلفة من الجدران والأسقف نبنـي عليهـا معرفتنـا الفيزيائيـة بطقـوس ومعالم النقل الحراري.

وتختلف الجدران عن الأسقف والبلاطات في درجة التعقيد ، لكنها في النهاية ليست إلا طبقات مختلفة تعبر الحرارة من خلالها.

Problem 1

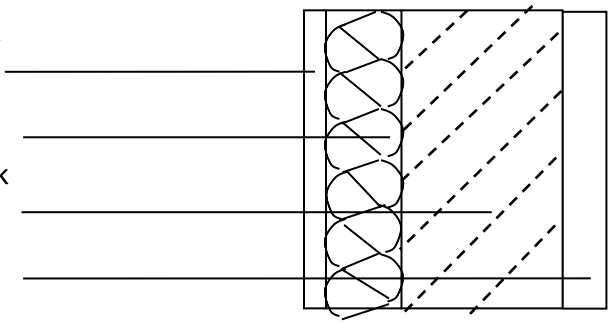
Please draw the thermal diagram for the following section and compare it with a transparent element regarding that the resistance is 0.14mk/w

0.5 cm , λ 0.2 w/mk

6 cm , λ 0.04 w/mk

24 cm , λ 0.99 w/mk

1,5 cm ,λ 0.2 w/mk



The air temperature inside is 20 ºc and outside is -15ºc

Solution

$$K = \left[\frac{1}{\alpha i} + \sum si/\lambda i + \frac{1}{\alpha o}\right]^{-1}$$

$$K_{wall} = \left[0.17 + \sum \frac{0.015}{0.7} + \frac{0.24}{0.99} + \frac{0.06}{0.04} + \frac{0.05}{0.7} + 0.04\right]^{-1}$$

K wall = <u>0.51 W/m²k</u>

$K_{glass} = [\,0.17 + 0.14 + 0.04]^{-1}$

$K_{glass} = \underline{2.86\ W/m^2k}$

$q_{wall} = kwall(\delta ai - \delta ao)$

$q_{wall} = 0.51\ (20 + 15)$

$q_{wall} = \underline{17.9\ W/m^2}$

$q_{glass} = 2.86\ (20 + 15)$

$q_{glass} = \underline{100.1\ W/m^2}$

يتضح من هذا أن التسرب الحراري مـن المـادة الشـفافة –الزجـاج- أكبـر بمقدار يزيد على الخمسة أضعاف من النفاذية عبر مادة مصمتة كالجدار مثلا. وذاك أمر منطقي. أما إذا أردنا حساب كمية الحـرارة ولـيس كثافـة الحرارة فعلينا ضرب الكثافة في المساحة ، وذاك هو الفرق بين Q وبـين q . بمعنى : $\underline{Q = q \cdot A\ [w]}$

To draw the thermal diagram we must calculate the temperature in different layers.

$$q = k\ (\delta ai - \delta ao) \qquad \text{........ 1}$$

$$q = [\frac{1}{\alpha i} + \sum si/\lambda i + \frac{1}{\alpha o}]^{-1}\ (\delta ai - \delta ao) \qquad \text{........ 2}$$

$$\delta Ii = \delta Ai - q\,\frac{1}{\alpha i} \qquad \text{.................3}$$

$$\delta 1 = \delta Ai - q\,[\frac{1}{\alpha i} + \frac{s1}{\lambda 1}] \qquad ....4$$

من المعادلات 3 ، 4 نستطيع حساب درجة الحرارة عند كل طبقة من طبقات الجدار . وبالإستتنتاج نستطيع أيضا معرفة المسافة داخل الجدار التي ستكون عندها درجة الحرارة تساوي كذا. يأتي هذا معنا لاحقا.

<u>The Temperature inside the wall</u>

$$\delta 1 = \delta Ai - q\,[\frac{1}{\alpha i} + \frac{s1}{\lambda 1}]$$

$$\delta 1 = 20 - 17.9\,[\,0.17 + \frac{0.015}{0.7}\,] = 16.6°c$$

$$\delta 2 = \delta Ao + q\,[\frac{1}{\alpha o} + \frac{s1}{\lambda 1}]$$

$$\delta 2 = -15 + 17.9\,[\,0.04 + \frac{0.06}{0.04}\,] = 12.6°c$$

$$\delta lo = \delta Ao + q\,[\frac{1}{\alpha o}]$$

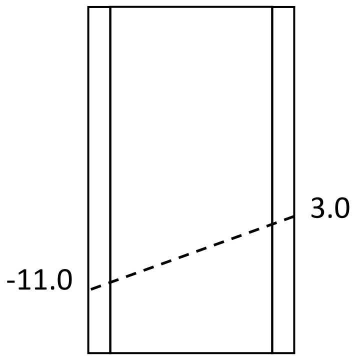
$$\delta lo = -15 + 17.9[0.04] = -14.3°c$$

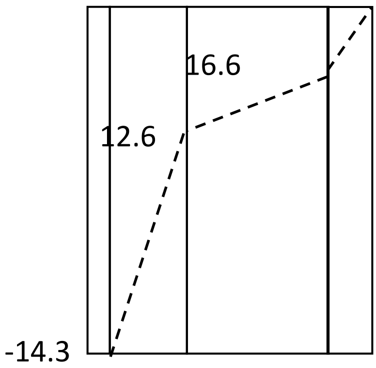
<u>The Temperature of the transparent part</u>

$$\delta li = \delta Ao - qglass\,[\frac{1}{\alpha i}] = 3.0°c$$

$$\delta lo = \delta Ao + qglass\,[\frac{1}{\alpha o}] = -11.0°c$$

Glass part diagram massive wall diagram





الرسم البياني يفهم منه سرعة إنتقال الحرارة في الجزء الشفاف ، فهو يمثل بخط مستقيم ، على عكس ما نجده في الجدار المصمت.

Problem 2

An office space, with the dimension 10x6x3 must be checked thermal Weiss. The façade include the layers:

1.5 cm cement , λ 0.7 w/mk

24 cm bricks , λ 0.99 w/mk

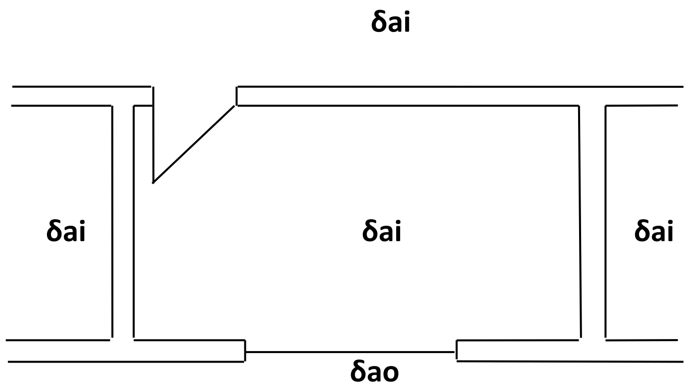
6 cm Insulation , λ 0.04 w/mk

Window:

	Type A	Type B
Frame	20%	20%
Transmission grad	0,8	0,4
Windows area	50%	50%

Conductivity 3.0 W/m^2k

Air-Heat capacity 0.35Wh/m^3k , Temperature difference $\Delta\ \delta$= 15k , Air change number n = 1.0 h^{-1}



- 6 person in the room with 120W heat convection from the human body for 6hour per day
- Artificial lighting 50 W/m^2 for 6hour per day
- Sun radiation on the window 30W/m^2 for 12 hour per day

Please calculate:

1-The average of the conductivity and the resistance for the total area of the façade, included the window area type A

2-The average of artificial Heating

3- Which conductivity for the window, if we don't want to heat artificially.

Solution

$$Km = \frac{K\ (window).A(window) + K\ (wall).A\ (wall)}{A\ (window) + A\ (wall)}$$

$$K = [\frac{1}{\alpha i} + \sum si/\lambda i + \frac{1}{\alpha o}]^{-1}$$

$$K\ wall = [0.13 + \sum \frac{0.015}{0.7} + \frac{0.24}{0.99} + \frac{0.06}{0.04} + 0.04]^{-1}$$

K wall = 0.52 W/m²k

Area of the wall

<u>A = 10 x 3 = 30m²</u>

Area of the window = 0.5 x Area = 0.5 x 30 = 15m²

Average of the total conductivity

$$K\ m = \frac{3 \times 15 + 0.52 \times 15}{30} = \underline{1.76\ w/m^2k}$$

The average of the total resistance

$$R = \frac{1}{km} - [\frac{1}{\alpha i} + \frac{1}{\alpha o}] = \underline{0.40 m^2k/w}$$

The artificial heating can be calculated through the principle of the thermal balance.

$$\boxed{\sum Q_{in} = \sum Q_{out}}$$

Q person = 6 x 120W x 6h/24h = 180W

Q art.lighting = 50 x 10 x 6 x 6h/24h = 750W

Q sun = I x A glass x τ x 12h/24h

A glass =A window x (1- r) = 15 x 0.8= $12m^2$

Q sun = 30 x 12 x 0.8 x 0.5 = 144 w

Q transmission= km . A . Δδ = 1.79 x 30 x 15 = 792 w

Q air = 0.35 x volum x n x Δδ = 0.35 x 10 x 6x 3x 1x15= 945w

Q heating = Q transmission + Q air –Q person –Q lighting–Q sun

= 663 w

Q heating = (k window. A window + k wall. A wall) Δδ + Q air –

Q person –Q lighting – Q window type B

Q window type B = 30 .12. 0.4 . 0.5 = 72 w

$$\textbf{K window type B} = \frac{\frac{Q\,heating - Q\,air + Q\,person + Q\,lighting + Q\,wB}{\Delta\delta} - k\,wall.A\,wall}{A\,window}$$

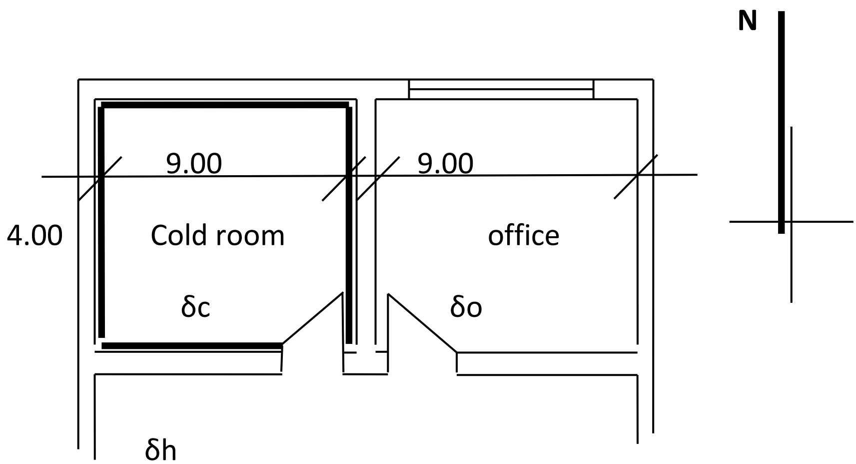
$$= \frac{1}{15}\left(\frac{663 - 945 + 180 + 750 + 72}{15} - 0.52.15\right) = \underline{2.7 w/m^2k}$$

إن هذا المثال الذي نحسب فيه مجموع الطاقات الحرارية الداخلية ، ومجموع الطاقات الحرارية الخارجية وهي الشمس ، ونساوي بين طرفي المعادلة بهدف الوصول إلى الفارق الحراري اللازم للتدفئة ،يمكن استخدامه أيضا من أجل التبريد .

كذلك يمكن حساب معامل التوصيل المناسب لزجاج النافذة للوصول إلى درجة حرارة معينة.

Problem 3

In a basement of one factory we have an office back to back to a cold room. The cold room has an insulation covering a round the walls. Please calculate the conductivity of the window and the average of the walls from the office. And which heating or cooling amount to have a constant temperature inside the space?



Giving

Volume dimension of each rooms

9.00 x 4.00 x 3.50m

Window 30% of the outside walls

Outside walls 24 cm concrete, λ 0.16w/mk

Inside walls 20 cm concrete, λ 0.21w/mk

Walls inclusive doors, R $0.75m^2k/w$

Roof R $1.0\ m^2k/w$

Flooring R $0.8\ m^2k/w$

Cold room insulation R 2 m^2k/w

Office façade Km 1.1 w/m^2k

Air-Heat capacity 0.35Wh/m^3k, Temperature outside = 20°c summer

Temperature outside = 0°c winter

Office temperature = 20°c

Cold room = 5°c

Production hall = 17°c

Earth temperature under the building= 10°c, Air change number n = 1.0 h^{-1}

Heat source inside office 2000W

Solution

Average of the conductivity exclusive wall and window

$$Km = \frac{K\ (window).A(window) + K\ (wall).A\ (wall)}{A\ (window) + A\ (wall)} = 1.1\ W/m^2k$$

Area window = 9 x 3.5 x 0.3 = 9.45 m^2

Area wall = 9 x 3.5 x 0.7 + 4 x 3.5 = 36.05 m^2

The total conductivity of the wall

$$K = [\frac{1}{\alpha i} + \sum si/\lambda i + \frac{1}{\alpha o}]^{-1} = [\ 0.13 + \frac{0.24}{0.16} + 0.04\ ]^{-1} = 0.6\ w/m^2k$$

The total conductivity of the window

$$K\ window\ =\ \frac{[Km,w+w\ (Area\ window+Area\ wall)-K\ (wall).A\ (wall)}{A\ (window)}$$

$$K\ window\ =\ \frac{[1.1\ x\ \ 45.5\ -0.6x\ 36.05]}{9.45} = 3.0\ W/m^2k$$

$$R\ m = \frac{1}{km} - [\frac{1}{\alpha i} + \frac{1}{\alpha o}] = \frac{1}{1.1} - 0.17 = 0.74\ m^2k/w$$

2) Artificial heating and cooling amount

∑ Qin = ∑ Q out

∑ Qheating **and/or** cooling = [$km_{wall+window}$. $A_{wall+window}$ + k_{roof} .A_{roof}+ 0.35. volume .n] (δoffice -δao) +k office, hall .A office, hall. (δoffice-δhall) + koffice, cold .A office, cold. (δoffice-δcold) + K earth .Aearth(δoffice -δearth) – Qin

K roof = $(0.13 + 1 + 0.04)^{-1}$ = 0.85 w/m²k

A roof = 9 . 4 = 36m²

Volume = 9.4.3,5= 126m³

K office, hall = $(0.13 + 0.75 + 0.13)^{-1}$ = 0.99 w/m²k

A office, hall = 9 . 3,5 = 31,5 m²

K office, cold = $(0.13 + \frac{0.2}{0.21} + 2 + 0.13)^{-1}$ = 0.31w/m²k

Aoffice, cold =4 . 3,5 = 14 m²

$Ke = (0.17+0.8)^{-1} = 1.03 W/m^2k$

$Ae = 9.4 = 36\ m^2$

∑ Qheating and cooling = [1.1 x 45.5 +0.85 x 36 + 0.35x126x1] [20-δao] + 0.99x 31.5 x3 + 0.31 x 14 x 15 + 1.03x 36 x 10 -2000 =

124.8 (20 –δao) – 1470.5

<u>We need the following energy for winter and sommer</u>

Winter ∑ Q= 1025.5 W ⟶ heating

Summer ∑ Q= - 1470.5 W ⟶ cooling

لـيس غريبـا أن يكـون للمهنـدس القـدرة وهـو فـي البـدايات التصـميمية لمشروع ما أن يعلـم مـن خـلال أبعـاد الفـراغ والظـروف المناخيـة كميـة الطاقة التي نحتاج إليها في التبريد صيفا والتدفئة شتاء.

وبنـاء علـى ماتقـدم لا نسـتطيع سـوى أن نسـلط الضـوء علـى أهميـة التصميم البيئي أو المعرفة والدراية بالعواقب الفيزيائيـة لفـراغ مـا إذا مـا كانـت معطياتـه كـذا وكـذا مـن حيـث الأبعـاد ومعـاملات التوصـيل وراحـة الإنسـان في داخله ، كل هذه المحددات تعطي قرارا تصميميا صائبا.

<u>الكوبرى الحرارى</u>

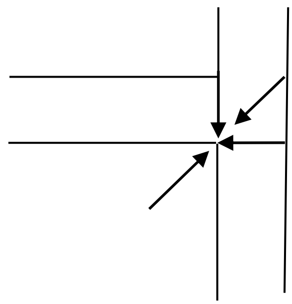
تبقـى مسـألة واحـدة وهـي أن الأركـان فـي المبـاني وإخـتلاف النظـام الإنشائي ومادة البناء من مسـببات إخـتلاف درجـة الحـرارة عنـد سـطح الجدار، الأمر الـذي يوصـف بالضـعف فيزئيـا لأن مـن توابعـه ظهـور النـدى والمياة الضحلة على سطح وفي ركن الحائط ، ويؤدي فـي النهايـة إلـى صدأ حديد التسليح إذا ما وصل إليه

Thermal Bridge

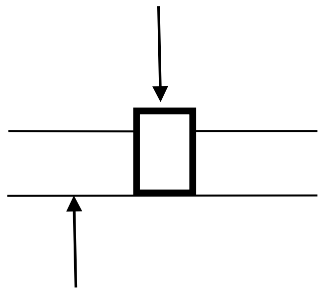
Thermal bridge is a phenomenon that expresses the wickniss thermaly of a wall. Different Temperature between the surface and the corner establisching the failer of the building. The thermal bridge has three types:

There are three types of thermal bridge:

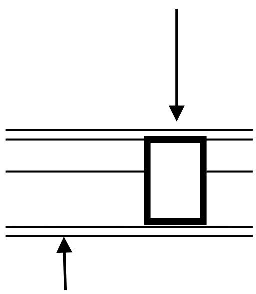
a) **Geometry**
The temperature in the corner is unequal with the temperature on the surface of the wall.



b) **Material**
The temperature on the masonry wall is unequal with the temperature from the surface of the concrete column.

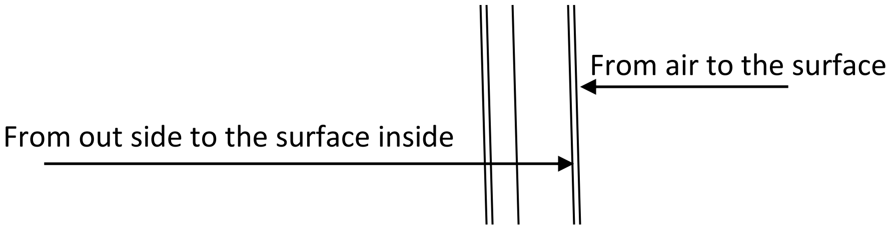


c) **Construction**
The temperature on the insulated part is unequal with the temperature in the non insulated part (column) .



Thermal Balance

The temperatur flow from inside is equal with the Temperature flow from outside. The calculation appears at any point of the layers. Finaly ∑ Qin = ∑ Q out



$$\mathbf{(\delta ai - \delta li)\left(\frac{1}{\alpha i}\right)^{-1} = (\delta li - \delta ao)\left(\frac{s1}{\lambda 1} + \frac{s2}{\lambda 2} + \frac{s3}{\lambda 3} + \frac{1}{\alpha o}\right)^{-1}}$$

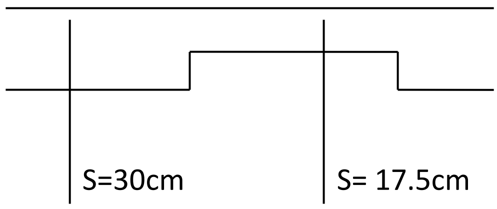
<u>We can calculate the thickness of the thermal insulation through the thermal balance.</u>

$$(\delta ai - \delta li)\left(\frac{1}{\alpha i}\right)^{-1} = (\delta li - \delta ao)\left(\frac{0.015}{0.7} + \frac{0.24}{0.99} + \frac{s}{0.04} + \frac{1}{\alpha o}\right)^{-1}$$

$$S = \left[\frac{1}{\alpha i} \cdot \frac{\delta li - \delta ao}{\delta ai - \delta li} - \left(\frac{0.015}{0.7} + \frac{0.24}{0.99} + \frac{1}{\alpha o}\right)^{-1}\right] 0.04 = 3.3 \cdot 10^{-3} m$$

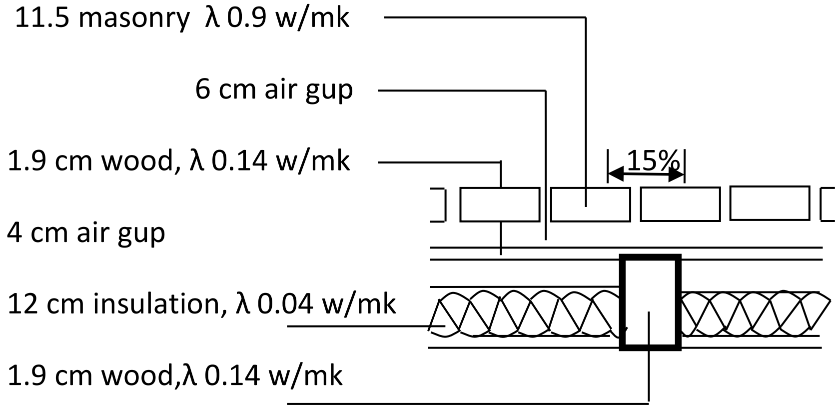
<u>= 0.3 cm</u>

The thermal bridge can appear by different thickness of the wall. See the sketch



Problem 4

- Please calculate the average of the heat transmission of the designed wall if the air gup conductivity λ = 0.17w/mk
- What is lost energy
- If we change the masonry wall with a façade 0.5cm material with λ 1.5 w/mk. How thick will be the insulation to have the same resistance like the masonry covering.



Temperature outside = 20°c summer

Temperature outside = 0°c winter

Solution

Average of the conductivity exclusive wall and window

$$K = [\frac{1}{\alpha i} + \sum si/\lambda i + \frac{1}{\alpha o}]^{-1}$$

$$= [\ 0.13+ \frac{0.019}{0.14}.2 + 0.17 + \frac{0.012}{0.004} + 0.17 + \frac{0.115}{0.96} + 0.04\ ]^{-1}$$

$$= \underline{0.26\ w/m^2}k$$

$Km = 0.6 \times 0.15 + 0.26 \times 0.85 = 0.31\ w/m^2k$

$$R = \frac{1}{km} - [\frac{1}{\alpha i} + \frac{1}{\alpha o}] = 3.1\ m^2k/w$$

The lost energy

$$Q = km\ .\ (\delta ai - \delta ao) = 6.2\ w/m^2$$

The new thickness of the insulation if we use the new material

$$KB \quad = [\ 0.13+ \frac{0.019}{0.14}\ .\ 2 + \frac{0.16}{0.17} + 0.08\ ]^{-1} = 0.7\ w/m^2k$$

The conductivity of the new material:

$$0.7 \times 0.15 + [\ 0.13+ \frac{0.019}{0.14}\ .\ 2 + 0.17 + \frac{s}{0.04} + 0.08\ ]^{-1}.\ 0.85 = 0.31$$

$S = 0.014m \longrightarrow = 2cm$

وبذا نستطيع ترشيد استخدام المواد بناء على كفاءتها للعزل، بمعنى أن اختيار المواد المستخدمة في الواجهات يكون بناء على ال Conductivity .

Problem 5

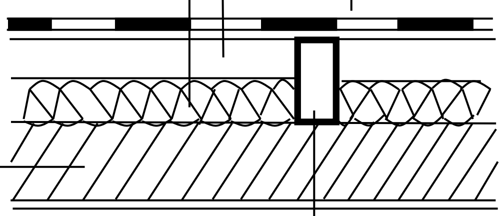
For the following roof construction, please calculate the following:

- How thick will be the air layer
- Where is the best location for the opening of the roof
- Please calculate r,q and δli for this roof.

Roof skin

Air insulation

10cm insulation λ 0.04w/mk, μ =1



16 cm RC , λ 2.1 w/mk, μ = 80

1.5 cm cement, λ 0.7 w/mk, μ = 10

Wood beam λ 0.17 w/mk

Note that μ is the Diffusion resistance number

Temperature outside = 20°c summer

Temperature outside = -10°c winter

Total area of the roof 100m²

Wood beams is 8% of the total area

Max length of the air layer is 12m

Solution

-Diffusion resistance $\sum(\mu.s) = 0.015x10 + 0.16x80 + 0.10x1 = 13m$

$$13\ m > 10m$$

The construction of the roof is from the aspect of the building physics is saved.

-The thickness of the air layer must be 5cm
-The location of the opening must be in front of each others to have ventilation and must be 2% from the total are of the roof.

The avarege of the resistance

$$R = \frac{1}{km} - [\frac{1}{\alpha i} + \frac{1}{\alpha o}]\ m^2k/w$$

$$Km = 0.08 \times kB + 0.92 \times KN \quad w/m^2k$$

$$K_B = [0.13 + \frac{0.015}{0.7} + \frac{0.16}{2.1} + \frac{0.10}{0.17} + 0.08\,]^{-1} = 1.12\ w/m^2k$$

$$K_N = [0.13 + \frac{0.015}{0.7} + \frac{0.16}{2.1} + \frac{0.10}{0.04} + 0.08\,]^{-1} = 0.36\ w/m^2k$$

$$\underline{Km = 0.08 \times 1.12 + 0.92 \times 0.36 = 0.42\ \ w/m^2k}$$

$$R = \frac{1}{km} - [\frac{1}{\alpha i} + \frac{1}{\alpha o}]\ m^2k/w$$

$R = \frac{1}{0.42} - [\,0.13+0.04] = 2.2\ m^2k/w$

Q = km .A. (δai – δao) = 0.42 x 100 x (20+10) = 1260 w

<u>The temperature of the surface in two different locations is:</u>

$\delta lo,B = \delta ai - kB\ (\delta ai - \delta ao)\ \frac{1}{\alpha i}\ = 15.6\ °c$

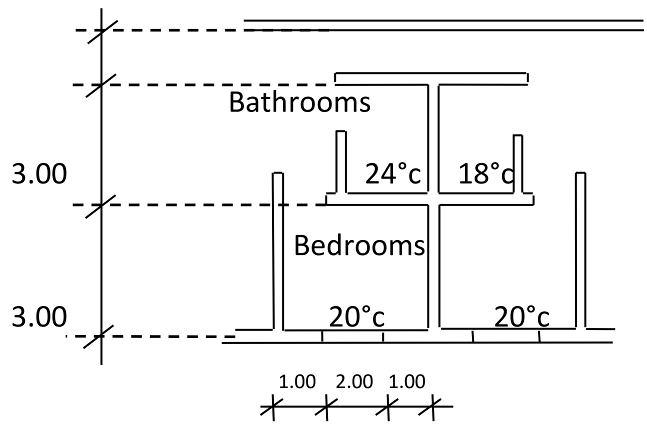
$\delta lo,N = \delta ai - kN\ (\delta ai - \delta ao)\ \frac{1}{\alpha i}\ = 18.6\ °c$

Problem 6

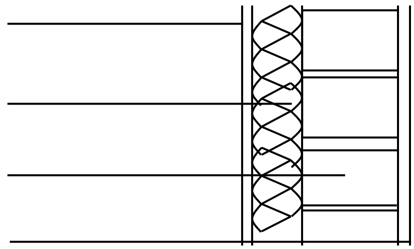
In a residential building we have two apartments, there bathrooms are placing back to back as shown. Please calculate:

- The conductivity and the resistance of the outside wall
- The average conductivity of the total façade
- In the Bedroom we have 500 w add value through artificial heating. Please calculate the air changing unit
- Calculate the temperature of the Bedroom on the surface of the façade if the radiation from the sun is 400w/m²

Sketch



Outer Wall



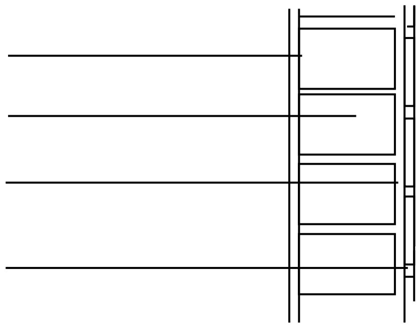
0.5cm cement,λ=0.70 w/mk

12.0cm insulation, λ=0.04 w/mk

24.0cm masonry,λ=0.99 w/mk

1.5cm cement,λ=0.70 w/mk

Inner Wall



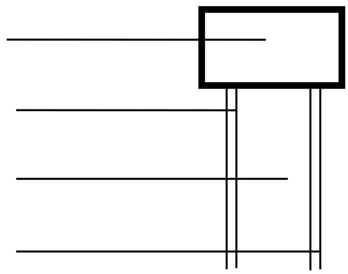
1.5 cm cement, λ = 0.70 w/mk

11.5cm masonry, λ = 0.99 w/mk

2cm cement x , λ = 1.4 w/mk

0.5 cm ceramic,λ = 1.0 w/mk

Window given data



5cm wood frame λ= 0.17 w/mk

0.3cm outside glass λ= 0.81 w/mk

1.2cm air between glass

0.3cm outside glass λ= 0.81 w/mk

Solution

The thermal resistance:

$$R = \frac{0.015}{0.7} + \frac{0.24}{0.99} + \frac{0.12}{0.04} = 3.26 \ m^2k/w$$

The thermal conductivity:

$$K = [0.13 + 3.26 + 0.04\,]^{-1} = 0.29 \text{ w/m}^2\text{k}$$

The average of the total façade

$$K\,m = \frac{kg.Ag + kf\,.Af}{Ag + Af} \quad [\text{w/m}^2\text{k}]$$

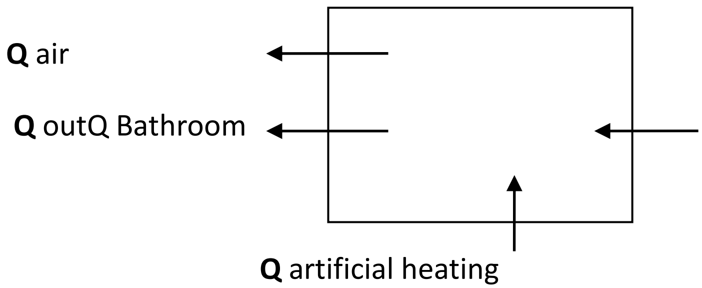
$$Kg = [0.13 + \frac{0.003}{0.81} + 0.14 + \frac{0.003}{0.81} + 0.04\,]^{-1} = 3.15 \text{ w/m}^2\text{k}$$

$$Kf = [0.13 + \frac{0.05}{0.17} + 0.04\,]^{-1} = 2.15 \text{ w/m}^2\text{k}$$

$$K\,m = \frac{kg.Ag + kf\,.Af}{Ag + Af} = 2.9 \;\; \text{w/m}^2\text{k}$$

The air changing number can we calculated through the thermal balance equations

$$\sum Q \;\; in \quad = \quad \sum Q \;\; out$$



Q Bathroom + **Q** artificial heating = **Q** out + **Q** air

Q bathroom= K b . Ab (δair bathroom –δair bedroom)

$$K_b = [\, 0.13 + \frac{0.015}{0.7} + \frac{0.115}{0.99} + \frac{0.02}{1.4} + \frac{0.005}{1.0} + 0.13 \,]^{-1} = 2.40 \text{ w/m}^2\text{k}$$

$A_b = 3 \times 2.5 = 7.5 \text{ m}^2$

Q bathroom = K_b . A_b (δ air bathroom – δair bedroom) = 72 W

Q heating = 500 W

Q wall = Q_{wall} + Q_{window} =198,9 W

Q out = Q wall + Q window

Q wall = Kw .Aw. (δair Bedroom – δ air outside)

Aw = 4 x 2.5 – Af = 10 – 2x1.35 = 7.3 m²

Q wall= 0.29 x 7.3 x 20 = 42.3 W

Q window =Kwindow .Awindow. (δair Bedroom – δ air outside)

Q window = 2.9 x 2.7 x 20 = 156.6 W

From the equation of the thermal balance

Q air = 500 + 72 – 198.9 = 373.1 W

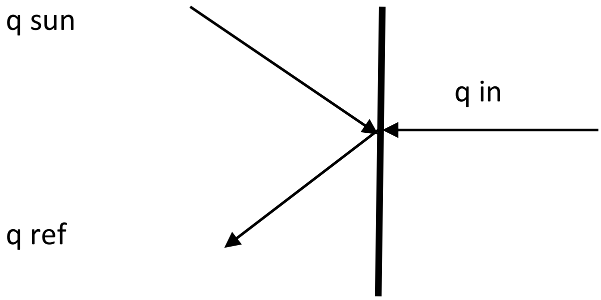
In the same time

Q air = n . ρ . c . volume. (δair Bedroom – δ air outside)

Volume = 3 x 4 x 2.5 = 30m²

n = $\frac{373.1}{0.35 \times 30 \times 2o}$ = 1.8 h^{-1}

The thermal balance on the façade is like the following sketch



with

Q to = qin + q sun, and Q out = q sun + q ref

Otherweis

<u>**q in + q sun = q ref**</u>

q in = k . (δ air bedroom – δ lo) = $[\frac{1}{\alpha i} + R]^{-1}$(δ air bedroom – δ lo)

= $(0.13 + 3.26)^{-1}$. (20 -δ lo)

q sun = I .a = 400 x 0.8 = <u>320 w/m²</u>

q ref = $\frac{\delta lo - \delta\ air\ outside}{\frac{1}{\alpha out}}$ = $\frac{1}{0.04}$ (δ lo – 0)

$$\frac{20-\delta lo}{0.13+3.26} + 320 = \frac{\delta\ lo}{0.04}$$

$$5.9 - 0.3 \times \delta lo + 320 = \frac{1}{0.04} + \delta lo$$

$$\delta lo = \frac{325.9}{25.3} = \underline{12.9°C}$$

هذا المثال مثال جيد جدا لمعرفة درجة حرارة سطح المبنى باسـتخدام معادلة الاتزان الحراري.

وفي هذا دليل واضح على أهمية تلك المعادلة ، ليس هذا فحسـب بـل بوسعنا أن نتعرف على سـمك طيقـة العـزل أو قيمـة معامـل الإمتصاص بإستخدام نفس المعادلة.

Problem 7

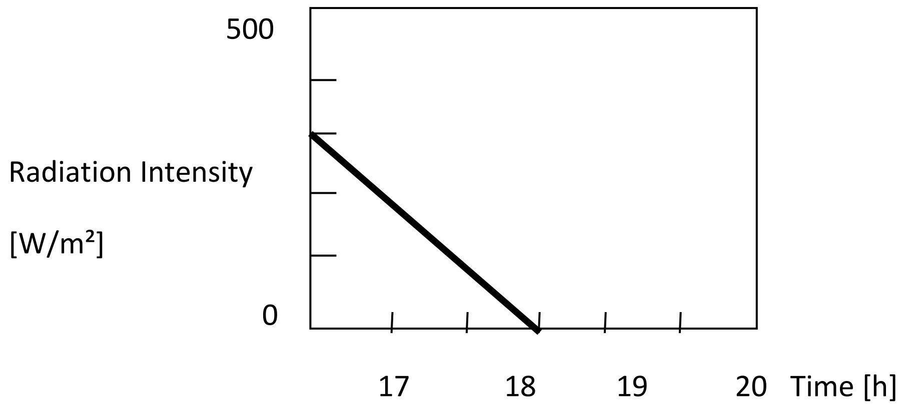
Calculate the artificial heating and draw the diagram in the time from 17 till 20 oclock for the outer wall of the building.

Given

Wall thickness	16cm
Wall area	12.5 m^2
Density	540kg/m^3
Conductivity	0.24 w/mk

Air temperature inside	20°C
Air temperature outside	-10°C

Energy resistance R inside	$0.13 m^2k/w$
Energy resistance R outside	$0.08 m^2k/w$



Specific heat capacity 0.28 Wh/kgk

Absorption grad 0.8

<u>Solution</u>

To calculate the heat period diagram we must calculate

1. First point $Si = \frac{\lambda}{\alpha i} = 0.13 \times 0.24 = 0.03m$

$$So = \frac{\lambda}{\alpha o} = 0.08 \times 0.24 = 0.02m$$

2. Layers number $\Delta x = \frac{d}{n} = \frac{0.16}{4} = 0.04$ m

Time distance grad $\Delta t = \frac{(\Delta x)^2}{2.a} = \frac{(\Delta x)^2}{2.\lambda}$ ρ. Cp [h]

$$\Delta t = \frac{(0.04)^2.540\ 0\ 0.28}{2.0.28} = 0.5\ h$$

Absorb temperature

$$\delta\ \textbf{absorb} = \delta ao + \frac{1}{\alpha o}\ I.a - k\ [°C]$$

which

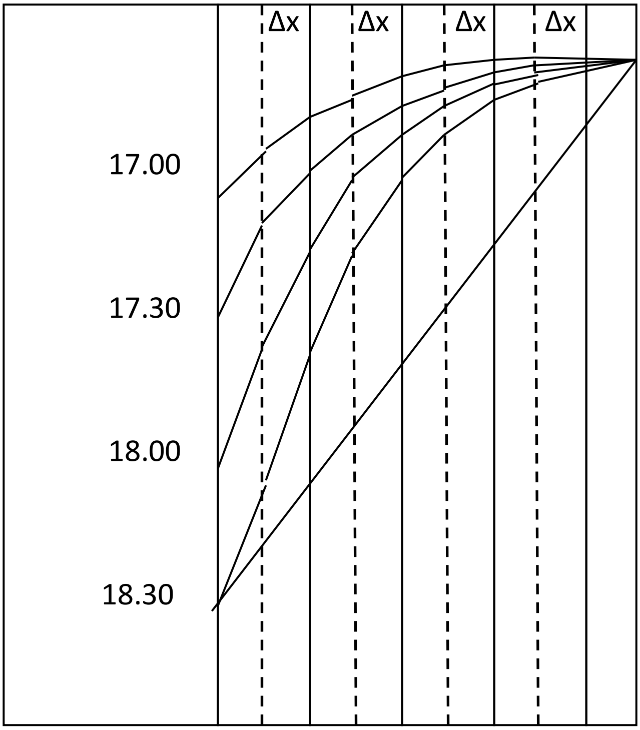
I = Intensity of radiation, a = absorption grad , K = corrector constant

Time [h]	17.00	17.30	18.00	18.30	19.00	19.30	20.00
1[w/m²]	300	200	100	0	0	0	0
K [k]	0	0	0	3	3	3	3
δabsorb	9.2	2.8	-3.6	-13	-13	-13	-13
δli	20	19.8	19.65	19.5	19.3	18.9	18.5
Q[W]	0	19.2	33.65	48	67.3	105.8	144.2

$$\mathbf{Q}_H = \frac{\delta ai - \delta li}{\frac{1}{\alpha i}} . A\ \ [W]$$

$$\mathbf{Q}_H = \frac{20 - 15.2}{0.13} .12.5 = 461.5\ [W]$$

The artificial heating amount is 461.5[w]



ماالذي نفهمه من هذا المثال؟

نفهـم أن معـدل التدفئـة المركزيـة يختلـف بسـبب إخـتلاف التسـرب الحراري، ولأن درجة الحرارة الخارجية متغيـرة ،فطبيعـي جـدا أن تختلـف الحرارة الداخلية على مدار الأربع وعشرين ساعة.

The Temperature at any layer of the wall

We can calculate the temperature in any point of the wall section through the following equations. Also we can determin the distance for some temperatue in the wall.

$$q = \frac{\lambda}{x} . (\delta ai - \delta x) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 1$$

$$\delta x = \delta ai - \left(\frac{x}{\lambda} . q \right) \text{ .... } 2$$

The total thermal mass

$$q = k . (\delta ai - \delta ao) \text{ .... } 3$$

$$= \left[\frac{1}{\alpha i} + \frac{s}{\lambda} + \frac{1}{\alpha o} \right]^{-1} (\delta ai - \delta ao) \text{ .... } 4$$

$$\delta x = \delta ai - \frac{x/\lambda}{\frac{1}{\alpha i} + \frac{s}{\lambda} + \frac{1}{\alpha o}} (\delta ai - \delta ao) \text{ .... } 5$$

$$x = \lambda \left[\frac{\delta ai \left(\frac{1}{\alpha i} + \frac{s}{\lambda} + \frac{1}{\alpha o} \right)}{(\delta ai - \delta ao)} \right] \text{ .... } 6$$

Conduction through cylinders

Conduction through cylinders can be calculated when variables such as the internal radius r1, the external radius r2 , and the length denoted as l . The temperature difference between the inner and outer wall can be expressed as T2 and T1

The area of the heat flow Ar = 2π. r . L

When Fourier's equation is applied:

$$\mathbf{Q = -K\ Ar} \frac{dt}{dr} \mathbf{= -2k.\pi.\ r.\ L} \frac{dT}{dr}$$

Rearranged:

$$\mathbf{Q} \int_{r1}^{r2} \frac{1}{r} dr \mathbf{= -2\ k\ \pi\ L} \int_{r1}^{r2} dt$$

Therefore the rate of heat transfer is :

$$\mathbf{Q = 2\ k\ \pi\ L} \frac{T1-T2}{\ln r2-\ln \quad r1}$$

$$R= \frac{\Delta T}{Q} = \frac{\ln r2-\ln r1}{2\pi.k.L}$$

Or ………………….

$$Q = 2\ k\ \boldsymbol{\pi}\ L.\ Rm\frac{T1-T2}{r2-\ r1}$$

$$Rm = \frac{r2-r1}{\ln r2-\ln r1}$$

Convection

Is the transfer of heat by the movement or flow of molecules (liquid or gas) with a change in their beat content? This is an important heat transfer mode between fluids and solids, or within fluids.

When a fluid such as water is heated, hotter portion of the fluid rises.

This is because as the mass of the fluid is heated, it expands. As the fluid expands it gets less dense compared to the surroundings and consequently rises.

Colder fluid takes the place of the warmer and a circulation takes place called circulation. The natural convection is called convection current. Some examples of natural convection are;

- The domestic hot water system inside the hot water cylinder
- The draught up a chimney
- The heating of rooms by convector heaters and radiators.
- The draught under a window.

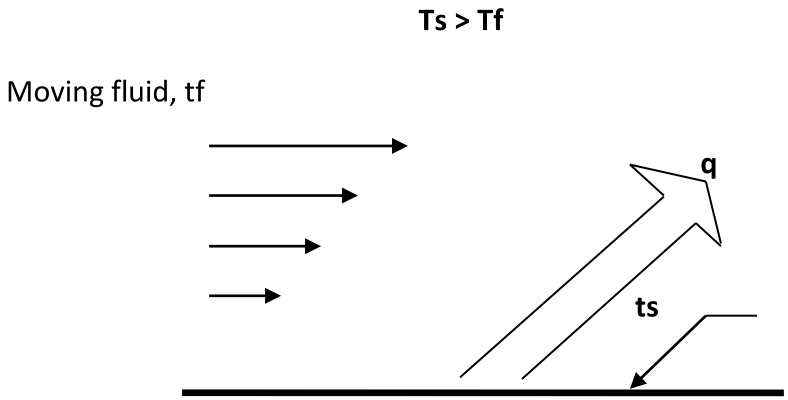


Fig:1.9 The heat transfer by convection process

Convective heat transfer is a mechanism of heat transfer occurring because of bulk motion (observable movement) of fluids (see convection for concept details). This can be contrasted with conductive heat transfer, which is the transfer of energy by vibrations at a molecular level through a solid or fluid, and radiative heat transfer, the transfer of energy through electromagnetic waves.

As convection is dependent on the bulk movement of a fluid it can only occur in liquids, gases and multiphase mixtures. Convective heat transfer is split into two categories: natural (or free) convection and forced (or advective) convection, also known as heat advection.

إن صورة النقـل الحـراري بطريـق الحمـل هـي الطريقـة الأكثـر شـيوعا . وتظهر بكثرة في إنبعاث الحرارة من القشرة الأرضية ، سواء كانت حـرارة أو تحول المياة إلى غاز من سطح القشرة الأرضية.

وبالمناسبة فالإحتباس الحراري أو زيادة حرارة الأرض بسـبب تـراكم غـاز ثاني أكسيد الكربـون عنـد قشـرة الغـلاف الجـوي ، هـي أحـد نـواتج ال CONNVECTION . وعليـه فإنعكـاس حـرارة الإشـعاع الشمسـي أمـر حتمي. ومما هو جـدير بالـذكر أن زيـادة حـرارة الأرض يتسـبب بهـا أيضـا إمتصاص رابطة الغازات الثلاثية الذرات ل IR وهو أمر يطـول شـرحه لكننـا ننوه إليه لكي يعلم القاصي والـداني أن الإحتبـاس الحـراري لـيس فقـط بسبب عدم تسرب حرارة ال CONNVECTION من الغلاف الجوي

ونقسم النقل الحراري بالحمل إلى

- النقل الحراري بالحمل حراً
- والنقل الحراري بالحمل عنوة

Onset of natural convection

Natural convection occurs when a system becomes unstable and therefore begins to mix by the movement of mass. A common observation of convection is of thermal convection in a pot of boiling water, in which the hot and less-dense water on the bottom layer moves upwards in plumes, and the cool and denser water near the top of the pot likewise sinks.

The onset of natural convection is determined by the Rayleigh number (Ra). This dimensionless number is given by

$$Ra = \frac{\Delta \rho g L^3}{D\mu}$$

Where

$\Delta\rho$ is the difference in density between the two parcels of Material that are mixing

G is the local gravitational acceleration

L is the characteristic length-scale of convection; the depth of the boiling pot, for example

D is the diffusivity of the characteristic that is causing the convection, and

μ is the dynamic viscosity.

Natural convection will be more likely and/or more rapid with a greater variation in density between the two fluids, a larger acceleration due to gravity that drives the convection, and/or a larger distance through the convecting medium. Convection will be less likely and/or less rapid with more rapid diffusion (thereby diffusing away the gradient that is causing the convection) and/or a more viscous (sticky) fluid.

For thermal convection due to heating from below, as described in the boiling pot above, the equation is modified for thermal expansion and thermal diffusivity. Density variations due to thermal expansion are given by:

$$\boldsymbol{\Delta\rho = \rho o.\ \alpha.\ \Delta T}$$

Where

ρo is the reference density, typically picked to be the average density of the medium,

α is the coefficient of the thermal expansion, and

ΔT is the temperature difference across the medium.

Inserting these substitutions produces a Rayleigh number that can be used to predict thermal convection.

The general diffusivity, D, is redefined as a thermal diffusivity, k.

D = k

$$\mathbf{Ra} = \frac{\rho o.g.\alpha.\Delta T.L^3}{k\mu}$$

Natural convective heat transfer

When heat is transferred by the circulation of fluids due to buoyancy from the density changes induced by heating itself, then the process is known as natural convection or free convection. Familiar examples are the upward flow of air due to a fire or hot object and the circulation of water in a pot that is heated from below.

For a visual experience of natural convection, a glass that is full of hot water filled with red food dye may be placed inside a fish tank with cold clear water. The convection currents of the red liquid will be seen to rise and also fall, then eventually settle, illustrating the process as heat gradients are dissipated.

Forced convection

Natural heat convection (also called free convection) is distinguished from various types of forced heat convection, which refer to heat advection by a fluid which is not due to the natural forces of buoyancy induced by heating.

In forced heat convection, transfer of heat is due to movement in the fluid which results from many other forces, such as (for example) a fan or pump. A convection oven thus works by forced convection, as a fan which rapidly circulation hot air forces heat into food faster than would naturally happen due to simple heating without the fan. Aerodynamic heating is a form of forced convection. Common fluid heat-radiator systems, and also heating and cooling of parts of the body by blood circulation, are other familiar examples of forced convection.

Flames and convection

In a zero-gravity environment, there can be no buoyancy forces and thus no natural (free) convection possible, so flames in many circumstances. Without gravity, smother in their own waste gases. However, flames may be maintained with any type of forced convection (breeze); or (in high oxygen environments in still gas environments) entirely from the minimal forced convection that occurs as heat-induced expansion (not buoyancy) of gases allows for ventilation of the flame, as waste gases move outward and cool, and fresh high-oxygen gas moves in to take up the low pressure zones created when flame-exhaust water condense.

ما الفرق بين النقل الحراري بالحمل حراً أو عنوةً

عندما تنتقل الحرارة عبر الحمل فإن ذلك يحدث عند دخول حرارة إلى سائل ما فتنتقل جزيئات السائل الساخنة إلى أعلى وتتحرك الجزيئات الباردة إلى الأسفل فيحدث النقل بالحمل عبر جزيئات السائل ويحدث أيضا داخل فراغ الغرفة في إنتقال الحرارة من سطح الجدار إلى هواء الغرفة.

إذا فإنتقال الحرارة بالحركة عبر السائل أو الهواء لإختلاف الكثافة يعرف بالحمل الحر. أما الحمل عنوة أو Forced Connvection فهي أيضا حركة للحرارة داخل جزيئات السائل لكن عبر مؤثرات خارجية. مثال ذلك المروحة أو المضخة.

أما بالنسبة للنقل الحراري بطريق الإشعاع فإنه يحدث في شكل تبادل لدرجات الحرارة بين مسطحين، تختلف درجة حرارتهما ، ويحدث ذلك في الجمادات والغازات دون إستثناء. وبالمناسبة فإن طبقات الشمس المختلتفة يوجد بها الصورتين من النقل الحراري ، الحمل والإشعاع ، وذلك في صورة Connvection zone, Radiative zone ، بمعنى أن النقل الحراري إما تبادلي أو عبر حركة الجزيئات بسبب إختلاف الكثافة.

<u>Radiation</u>

Heat transfer by this mode therefore requires a line of sight connection between the surfaces involved. All objects above absolute zero radiate heat energy; it is the net radiative heat transfer that is the heat transfer of interest. Radiation is mostly of importance for heat transfer between solids and within highly porous solids, but

radiation between high-temperature gases in occasionally of practical importance. Heat can transferred through a vacuum from one body to another. A good example of this is the heat from the sun, which passes through space to reach the earth.

This process is neither convection nor conduction. The heat is carried in wave form in the same way as light is carried. This is called electromagnetic radiation. Other forms of electromagnetic radiation are infrared, ultra-violet, x-rays and gamma rays.

As heat radiation is part of the same spectrum as light radiation, it obeys approximately the same rules, the most important ones being:

-It does not require a medium for transmission
-It travels in straight lines.
-It can be reflected and reflected in the same way as light.

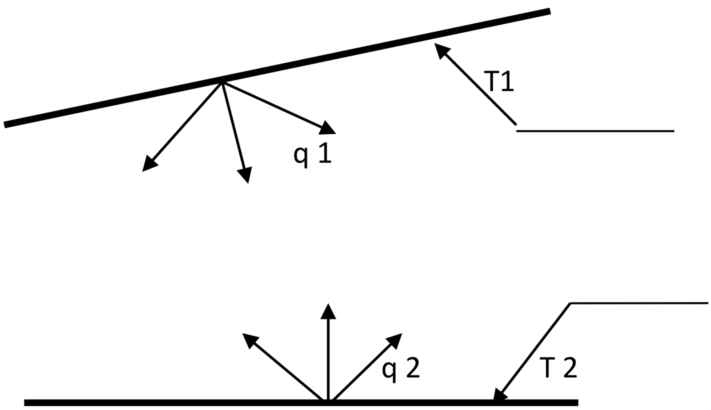
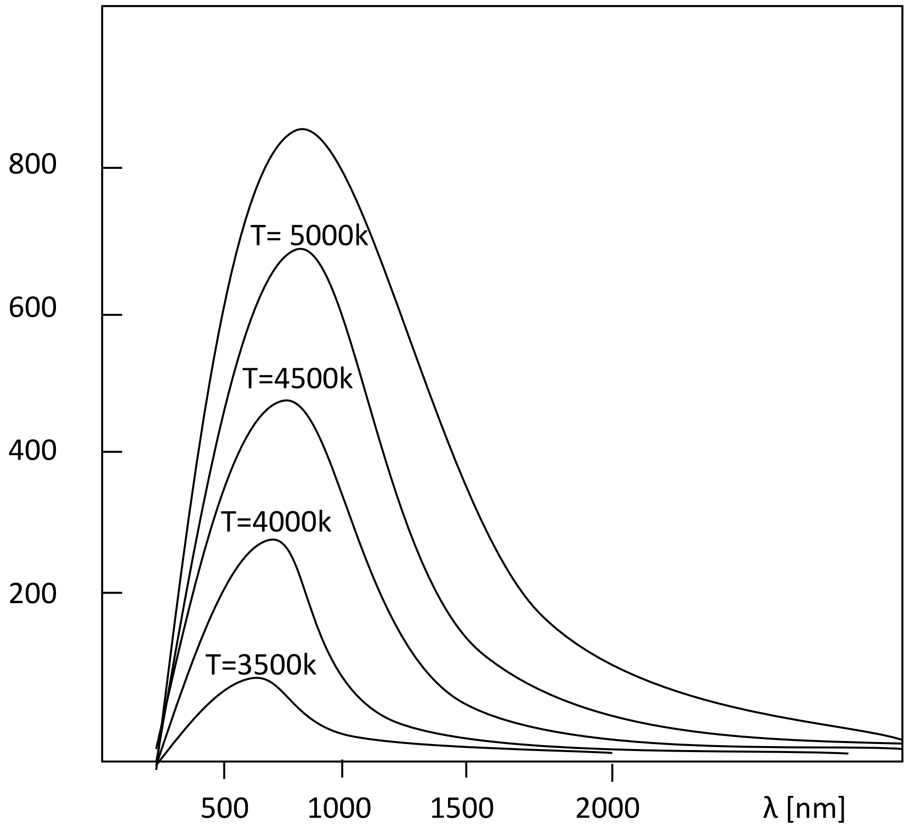


Fig:1.10 The heat transfer by radiation process



This diagram shows how the peak wavelength and total radiated amount vary with temperature. Although this plot shows relatively high temperatures, the same relationships hold true for any temperature down to absolute zero. Visible light is between 380 to 750nm

Properties

There are three main properties that characterize thermal radiation:

-Thermal radiation, even at a single temperature, occurs at a wide range of frequencies. How much of each frequency is

given by plank's law of radiation (for idealized materials). This is shown by the curves in the diagram at right.

-The main frequency (or color) of the emitted radiation increases as the temperature increases. For example, a red hot object radiates most in the long wavelengths of the visible band, which is why it appears red. If it heats up further, the main frequency shifts to the middle of the visible band, and the spread of frequencies mentioned in the first point make it appear white. We then say the object is white hot. This is Wien's displacement law. In the diagram the peak value for each curve moves to the left as the temperature increases.

-The total amount of radiation, of all frequencies, goes up very fast as temperature rises (it grows as T^4, where T is the absolute temperature of the body). An object at the temperature of a kitchen oven (about twice room temperature in absolute terms – 600k vs. 300k) radiates 16 times as much power per unit area. An object at the temperature of the filament in an incandescent bulb (roughly 3000k, or 10 times room temperature) radiates 10.000 times as much per unit area. Mathematically, the total power radiated rises as the fourth power of the absolute temperature, the Stefan-Bolzmann law. In the plot, the area under each curve rises rapidly as the temperature increases.

These properties apply if the distances considered are much larger than the wave lengths contributing to the spectrum (around 10 Micrometres at 300k). Indeed, thermal radiation here takes only travelling waves into account. A more sophisticated frame work involving electromagnetic has to be used for lower distance (near- field thermal radiation).

Interchange of energy

The thermal radiation is an important concept in thermodynamics as it is partially responsible for heat exchange between objects, as warmer bodies radiate more heat than colder ones (other factors are convection and conduction). The interplay of energy exchange is characterized by the following equation:

$$\alpha + \rho + T = 1$$

Here, α represent spectral absorption factor, ρ spectral reflection factor and T spectral transmission factor.

This entire element depends also on the wavelength λ. The spectral absorption factor is equal to the emissivity Є, this relation is known as Kirchhoff's law of thermal radiation. An object is called a black body if, for all frequencies the following formula applies;

$$\alpha = \varepsilon = 1$$

In a practical situation and room-temperature setting, humans lose considerable energy due to thermal radiation. However, the energy lost by emitting infrared heat is partially regained by absorbing the heat of surrounding objects (the remainder resulting from generated heat through metabolism). Human skin has an emissivity of very close 10 1.0 using the formulas below then shows a human being, roughly 2 square meter in area, and about 307 kelvins in temperature, continuously radiates about 1000 watts. However, if people are indoors, surrounded by surfaces at 296 k, they receive back about 900 watts from the wall, ceiling, and other surroundings, so the net loss is only about 100watts. These heat transfer estimates are highly depended on extrinsic variables, such as wearing clothes (decreasing total thermal circuit conductivity, therefore reducing total output

heat flux) only truly grey systems relative equivalent emissivity and no directional transmissivity dependence in all control volume bodies considered . Can achieve reasonable irradiative flux estimates through the Stefan-Bolzmann law. However encountering this ideally calculable situation is virtually impossible although common engineering procedures surrender the dependency of these unknown variables and assume this to be the case. Optimistically, these grey approximations will get you close to real solutions, as most divergence from Stefan-Bolzmann solutions is small (especially un most lab controlled environments).

If objects appear white (reflective in the visual spectrum), they are not necessarily equally reflective (and thus non-emissive) in the thermal infrared; e.g. most household radiators are painted white despite the fact that they have to be good thermal radiators. Acrylic and urethane based white paints have 93% black body radiation efficiency at room temperature (meaning the term black body does not always correspond to the visually perceived color of an object). These materials that do not follow the "black color = high emissivity/absorptivity " caveat will most likely have functional spectral emissivity/absorptivity dependence.

Calculation of radiative heat transfer between groups of object, including a cavity or surroundings requires solution of a set of simultaneous equations using the radiosity method. In these calculations, the geometrical configuration of the problem is distilled to a setoff numbers called view factors, which give the proportion of radiation leaving any given surface that hits another specific surface.

These calculations are important in the fields of solar thermal energy, boiler and furnace design and ray traced computer graphics.

Formula

Thermal radiation power of a black body per unit of area, unit of solid angle and unit of frequency v is given by plank's law as:

$$\mathbf{U(v,T)} = \frac{2hv^3}{c^2} \cdot \frac{1}{e^{\frac{hv}{k_B T}} - 1}$$

This formula mathematically follows from calculation of spectral distribution of energy in quantized electromagnetic field which is in complete thermal equilibrium with the radiating object.

$$\mathbf{W = \sigma \cdot A} \cdot T^4$$

Where the constant of proportionality σ is the Stefan-Bolzmann constant and A is the radiating surface area.

Further, the wavelength λ for which the emission intensity is highest, is given by wien's law as:

$$\boldsymbol{\lambda}\mathbf{max} = \frac{b}{T}$$

For surface which are not black bodies, one has to consider the (generally frequency dependent) emissivity correction factor ε(u). This correction factor has to be multiplied with the radiation spectrum formula before integration. The resulting formula for the

power output can be written in a way that contains a temperature dependent correction factor which is (somewhat confusingly) often called ε as well:

$$W = \varepsilon(T).\sigma.A.T^4$$

Constants

h	Planck's constant	$6.626.\ 10^{-34}$ J.s = $4.135.10^{-15}$ eV.s
b	wien's displacement constant	$2.897.10^{-3}$ mk
kB	Boltzmann constant	$1.38.\ 10^{-23}$ Jk^{-1} = $8.61\ .10^{-5}$ eV.s
σ	Stefan-Bolzmann constant	$5.670\ .10^{-8}$W.$m^{-2}k^{-4}$
c	Speed of light	299.792. 458 m.s^{-1}

Radiations law

$$M = C_1\ .\ \frac{\lambda^{-5}}{(\exp\frac{c_2}{\lambda.T}-1)}$$

$$C_1 = 2\pi.c^2.h$$

$$C_2 = c.h/k$$

Which c is the speed of light, h is Planck's constant and k is Boltzmann constant

$$Ms = \sigma . T^4$$

$$T = \delta + 273.16$$

$$Ms = Cs \, . \left(\frac{T}{100} \right)^4$$

Which Cs is the radiation constant of the black body .

We have three types of rays : reflected rays , absorption rays and entered rays.

$$\rho = \frac{reflection\ rays}{total\ radiation}$$

$$\alpha = \frac{absorpation\ rays}{total\ radiation}$$

$$\tau = \frac{entered\ rays}{total\ radiation}$$

$$\tau + \alpha + \rho = 1$$

The diagram shows us the relation between the maximum of radiation and the temperature. From the Planck law of radiation we can understand that with the temperature the short wave is the result.

$$\lambda \max \ .T = 2896 \ . \ 10^{-6} \ mk$$

Emission and Absorption

The spectral specific radiation of a temperature is from black body radiation defined. The emission grad is the dividing of the emission grad of the radiated body on the emission grad of the black body.

$$\varepsilon = \frac{M}{Ms}$$

Similar to this equation

$$\varepsilon = \frac{C}{Cs}$$

Which ….

C = radiation constant of any body

Cs = radiation constant of black body

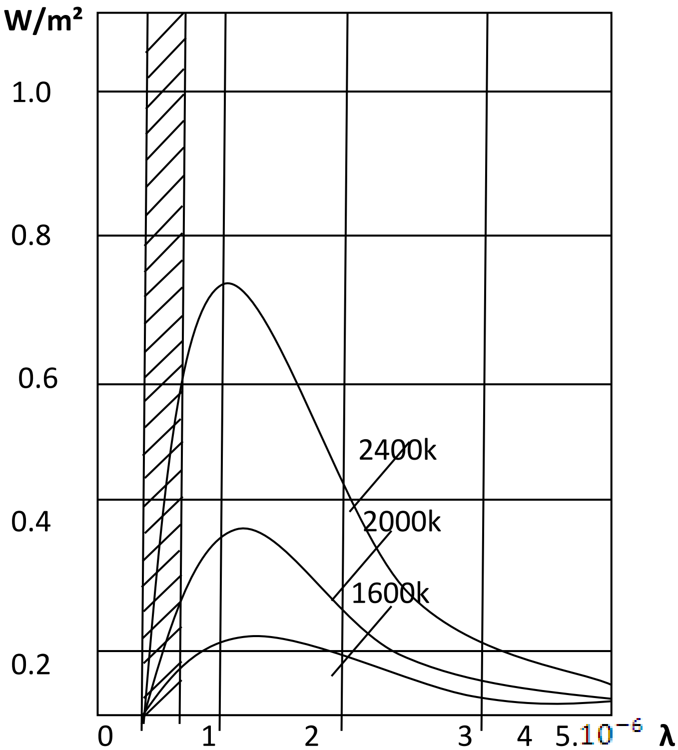


Fig:1.11 The relation between the maximum of radiation and the temperature

The specific radiation M :

$$M = \varepsilon \cdot \sigma \cdot T^4$$

$$M = \varepsilon \cdot Cs \left(\frac{T}{100} \right)^4$$

Radiation between parallel surfaces

$$Q = C_{1,2} \cdot A \cdot \left[\left(T1/100 \right)^4 - \left(T2/100 \right)^4 \right]$$

$$C_{1,2} = \frac{Cs}{\frac{1}{\varepsilon 1} + \frac{1}{\varepsilon 2} - 1}$$

$$a = \frac{\left(\frac{T1}{100}\right)^4 - \left(\frac{T2}{100}\right)^4}{T1 - T2}$$

$$\alpha r = a \cdot \frac{Cs}{\frac{1}{\varepsilon 1} + \frac{1}{\varepsilon 2} - 1}$$

$$Q = \alpha r \cdot A \cdot (T1 - T2)$$

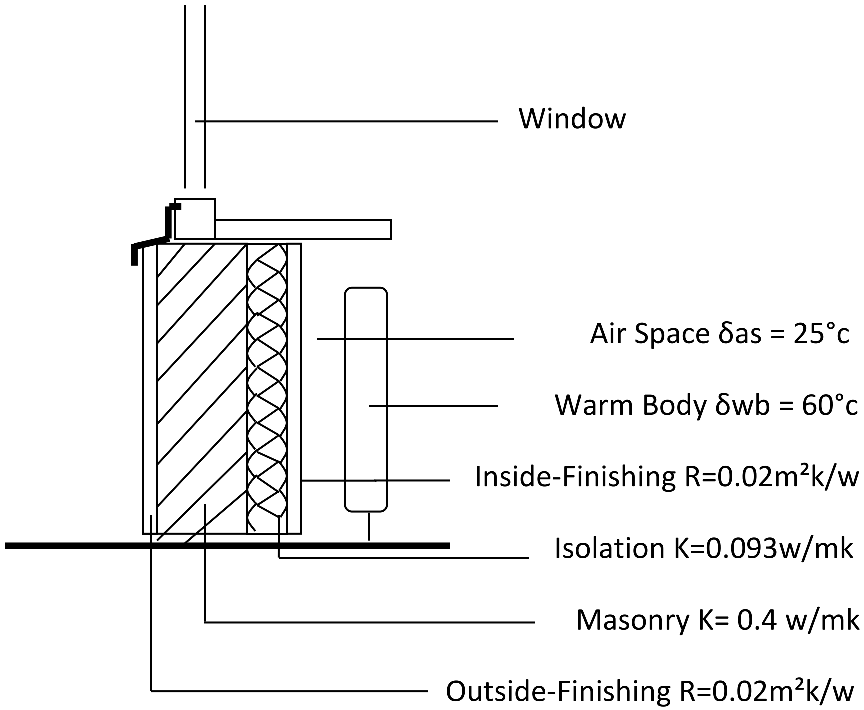
$$Q = \alpha r \cdot A \cdot (\delta 1 - \delta 2)$$

The Conductivity radiation factor αr is different from material to another. Tab show us the difference between metal and nonmetals factors.

Surface Combination	Temperature δ1 in °c	αr by δ2 -10	αr by δ2 10	αr by δ2 30	αr by δ2 50
A	-10 10 30 50	3.4 3.8 4.2 4.7	3.8 4.2 4.7 5.2	4.2 4.7 5.2 5.7	4.7 5.2 5.7 6.3

B	-10 10 30 50	0.21 0.23 0.26 0.29	0.23 0.26 0.28 0.32	0.26 0.28 0.31 0.35	0.29 0.32 0.35 0.38
C	-10 10 30 50	0.11 0.12 0.13 0.15	0.12 0.13 0.15 0.16	0.13 0.15 0.16 0.18	0.15 0.16 0.18 0.20

Problem 8



Outside wall

Inside Finshing	R= $0.02 m^2 k/w$

30cm Masonry wall	λ = 0.4W/mk
Outside-Finishing	R= 0.02m²k/w
Window frame and glaspart	K= 3.0 w/m²k

Air-Temperature Inside 20°c

Outside 0°c

Radiation constant of black body Cs = 5.77 W/m²k^4

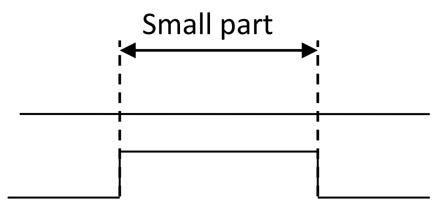
Conductivity of the surface of the wall from inside 0,17 m²k/w

Conductivity of the air between the wall and warm body 0,13 m²k/w

- Calculate the thickness of the isolation in the small part of the wall
- Calculate the lost energy in the small part of the wall
- The radiation between the two parallels.
- Calculate the emission number of the warm body
- Calculate the total lost energy

Solution

The isolation layer of the wall in the small part



Wall $\quad R = 0.02 + \frac{0.3}{0.4} + 0.02 = 0.79\ m^2k/w$

Small part $\quad R = 0.02 + \frac{0.175}{0.4} + \frac{s}{0.093} + 0.02 = 0.79\ m^2k/w$

$\longrightarrow$ S = 0.029m = 3cm

<u>Heat balance</u>

1) qs= qb + qw

$(\frac{1}{\alpha s})^{-1}.(\delta b - \boxed{\delta li}) = (\frac{1}{\alpha b})^{-1}.(\delta li - \delta as) + (\frac{1}{\alpha o} + R)(\delta li - \delta ao)$

$(0.17)^{-1}.(60 - \delta li) - (0.13)^{-1}.(\delta li - 25) - (0.04 + 0.79)^{-1}.(\delta li - 0) = 0$

$\delta li = 36.9°c$

<u>2) To calculate the lost energy in the small part we must calculate both part of the wall.</u>

q small part = $(0.04 + 0.79)^{-1}.(36.9 - 0) =$ <u>44.5 w/m²</u>

q wall = $(0.13 + 0.79 + 0.04)^{-1}.(20 - 0) =$ <u>20.8 w/m²</u>

Lost Energy = $\frac{44.5-20.8}{20.8}$ = 1.14 = 114 % higher than the rest of the wall

3) The radiation intensity

$$\alpha s = C1{,}2 \ . \ \frac{(\frac{Tb}{100})^4 - (\frac{Tli}{100})^4}{\delta b - \delta li}$$

$$C1,2 = \frac{1}{0.17} \cdot \frac{60-36.9}{(\frac{273+60}{100})^4 - (\frac{273+36.9}{100})^4} = 4.42\ w/m^2 k^4$$

4) The Emission number of the warm body

$$C1,2 = \frac{Cs}{\frac{1}{\varepsilon 1}+\frac{1}{\varepsilon 2}-1} \quad [w/m^2 k^4]$$

$$E1 = (\frac{5.77}{4.42} + 1 - \frac{1}{0.9})^{-1} = 0.34$$

5) The total lost energy

qt = 0.2 q small part + 0.5 q wall + 0.3 q window

q small part = 44.5 w/m^2

q wall = kw (δai –δao)=20.8 w/m^2

q wand = kw (δai –δao)= 3. (20-0) = 60 w/m^2

q total = 0.2 x 44.5 + 0.5x 20.8 + 0.3 x 60 = 37.3 w/m^2

يلاحظ مـن المثـال السـابق أننـا فـي الفقـرة الأولـى اسـتخدمنا معادلـة الإتزان الحراري للحصول على درجـة حـرارة سـطح الحـائط δli . والجـدير بالذكر أن شرط هذه المعادلة تصفيرها عند أي نقطة من نقـاط الحـائط ، بمعنى أنه جائز جدا أن تكون مجموع القـوى يسـاوي صـفر عنـد سـطح الحائط أو عند أي طبقة من الطبقات δ1..δ2

ومن خلال درجة الحرارة اسـتطعنا أن نحسـب كميـة الحـرارة المتسـربة من خلال جزء الجدار النحيل أمام التدفئة المركزية وجـزء الجـدار العـادي ونحدد بعدها الفقد الحراري عبر الجدار.

Problem 9

The flat roof of a hospital includes the following layers.

1cm Cement	λ 0.07 w/mk
20cm concrete	λ 2.1 w/mk
Isolation	λ 0.04 w/mk

Temperature	Inside	20°c
	Outside	0°c

Calculate

-How thick is the isolation to have only 6w/m² heattransnission?

-In case that we have a warm body in the flat roof of the hospital with heat intensity 60.4 w/m² how thick is the isolation?

Solution

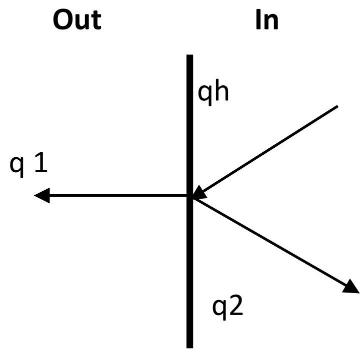
$K = \frac{q}{\delta ai - \delta ao} = \frac{6}{20-0} = 0.3 \text{ w/m}^2\text{k}$

$R = \frac{1}{k} - \left(\frac{1}{\alpha i} + \frac{1}{\alpha o} \right) = 3.2 \text{ m}^2\text{k/w}$

$R = \sum \frac{si}{\lambda i} = \frac{0.01}{0.7} + \frac{0.2}{2.1} + \frac{S}{0.04}$

$S = 0.04 \left(3.2 - \frac{0.01}{0.7} - \frac{0.2}{2.1} \right) = 0.04 \times 3.09 = \underline{0.124 \text{ m}}$

2) to calculate the thickness we must know another form from the heat balance.



$\sum$ q in = $\sum$ q out

<u>q h = q1 + q2</u>

q Warm body = q out side + q inside

q H = 60.4 w/m^2

$q1 = (\frac{1}{\alpha i})^{-1}. (\delta li - \delta ai)$ $[w/m^2]$

$q2 = (R+ \frac{1}{\alpha o})^{-1}.(\delta li - \delta ao)$ = $6 w/m^2$

$60.4\ w/m^2 = (\frac{1}{\alpha i})^{-1}.(\delta li - \delta ai) + 6\ w/m^2$

The temperature of the level outside of the wall is :

$\delta lo = \frac{1}{\alpha i} . 54.4 + \delta ai = 5.44 + 20 = 25.4$ °c

The resistance of the wall R

R = $\frac{\delta li - \delta ao}{6}$ - $\frac{1}{\alpha o}$ = $\frac{25.4}{6}$ - 0.04 = 4.2 m^2k/w (This resistance is in case of using the warm body)

R new = R with warm body – R without Warm body

R new = 4.2m^2k/w - 3.2 m^2k/w = 1 m^2k/w

<u>The thickness of the insulation can by about 33% higher than before</u>

<u>Problem 10</u>

A room with isolated glass windows has the same air change number from the other rooms. The exchange of air exists through windows.

Room Dimension 5 x 5 x 2.5

1.5cm cement	λ = 0.7 w/mk
2.5cm cement	λ = 0.87 w/mk
Glass window 2.5 m^2	k = 3.0 w/m^2k

Outside wall 36.5 cm masonry

Air capacity of heat 0.35 $wh/m^3 k$

Air Temperature Inside 20°c

Air Temperature outside 0°c

Air change number $1.0h^{-1}$

<u>Calculate</u>

-The Thermal Conductivity of the wall

-The artificial heating

<u>Solution</u>

$$\textbf{Km} = \frac{kwindow\ .Awindow\ +\ k\,wall\ .A\,wall}{A\ window + A\ wall} \quad \textbf{[w/m}^2\textbf{k]}$$

$$K\ wall = \frac{km\ (Awindow + A\ wall) - K\ window\ .A\ window}{A\ wall}$$

$$K\ wall = \frac{1.16\ .5\ .2{,}5 - 3\ .2{,}5}{5\ .2.5 - 2{,}5} = 0.7\ w/\ m^2k$$

$$R = \frac{Sm}{\lambda m} = \frac{1}{0.7} - 0.13 - \frac{0.015}{0.7} - \frac{0.025}{0.87} - 0.04 = 1.21\ m^2k/w$$

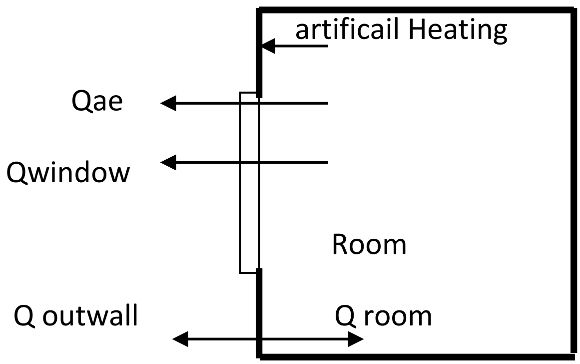
$$\lambda m = \frac{0.365}{1.21} = \underline{0.30\ w/mk}$$

الأمر الهام في هـذه الجزئيـة أننـا إذا حـددنا معامـل التوصـيل المناسـب لدرجة حرارة داخلية مرغوبة ، إستطعنا أيضا معرفة اسـم تلك المادة من جداول معاملات التوصيل .

وبالتالي أصبح كل ما نقدمه من حسـابات يهـدف ضـمن مـا يهـدف إلـى اختيار المواد الملائمة للفراغات الداخلية المرجوه.

-The artificial heating

$$\sum q_{in} = \sum q_{out}$$



Q room = Q window + Q air exchange

Q window = K window. Area window (δ ai – δao)

= 3.0 x 2.5 x (20-0) = 150 w

Q air exchange = ne . ρe. Ce .V (δai-δao)

= 1.0 x 0.35 x 5 x 5 x 2.5 x20 = 437.5W

Q artificial heating = Q room + Q wall

Q room = 587.5 W

Q room = $(\frac{1}{\alpha i})^{-1}$. A wall (δli –δai)

δli= δai+ $\frac{Q\ room}{A\ wall\ .\alpha i}$= 20+0.13 x 58.75 = 27.6°c

Q wall = $(\frac{1}{kw} - \frac{1}{\alpha i})^{-1}$ .Awall (δli – δao)

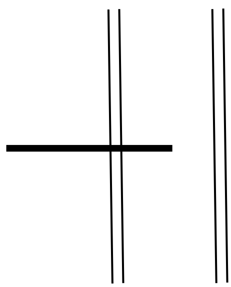
= $(\frac{1}{0.7}$ -0.13$)^{-1}$x 10 x (27.6 – 0 = 212.5 W

Q artificial heating = 587.5 + 212.5 = 800W

Problem 11

The glass window has the following data.

C1,2 = 5w/$m^2 k^4$



The thermal conductivity in the space between the two glass levels 3.1w/m²k

Glass thickness 4mm

Conductivity 0.8w/mk

Space in between 10 mm

Air Temperature Inside 20°c

Air Temperature outside 0°c

Calculate

The thermal intensity of the space between the two glass levels

Solution

$K = (0.13+ \frac{0.004}{0.8} + 0.14 + \frac{0.004}{0.8} + 0.04)^{-1} = 3.1$ w/m²k

$Q\ s = C1,2 . [(\frac{T1}{100})^4 - (\frac{T2}{100})^4]$

$\delta li = 20 - \frac{1}{\alpha i} . q = 20 – (0.13 + \frac{0.004}{0.8}) .62 = \underline{11.6°c}$

$\delta lo = 0 + \frac{1}{\alpha o} . q = 0 + (\frac{0.004}{0.8} + 0.04) .62 = \underline{2.8°c}$

$T1 = \delta li + 273 = 284.6k$

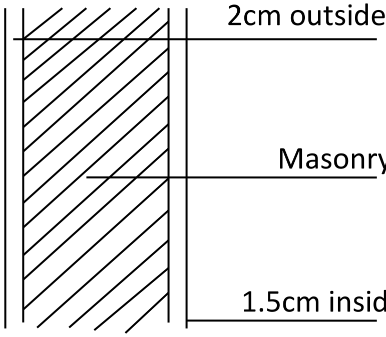
$T2 = \delta lo + 273 = 275.8k$

$qs = 5.(2.846^4 - 2.758^4) = 38.7$ W/m²

q glass levels = 62 – 38.7 = <u>23.3 W/m²</u>

Problem 12

The elevation of a megastore building includes glass area of 40% of the total façade. The frame of the glass area is about 20%. All others wall have a temperature from 20°c. The design of the elevations wall is as following:



Room Dimension 4 x 4 x 2.5

Elevation area 10m²

Window 4mm Glass λ 0.8 w/mk

Window frame 6.8cm from wood λ 0.13 w/mk

Air changing number 0.35wh/m^3 k

Temperature ………….. inside 20°c

(Night) Outside 0°c

Day average 5°c

Solar Intensity (day average) 250w/m²

Calculate

-The thickness of the wall to have a resistance of $0.55m^2$ k/w

-The heat transmission of the window

-The heat transmission of the wall in the night and in the day average temperature if the reflection grade of the color of the façade 0.5

-The artificial heating

Solution

The Thickness of the wall

$$\sum \frac{si}{\lambda i} = \frac{0.015}{0.7} + \frac{sm}{0.72} + \frac{0.02}{0.87} = 0.55m^2k/w$$

$$Sm = 0.364 \text{ m} \longrightarrow 36.4\text{cm}$$

The heat transmission of the window

$$K= \left[\frac{1}{\alpha i} + \frac{s}{\lambda} + \frac{1}{\alpha o}\right]^{-1} = \left[0.13 + \frac{0.004}{0.8} + 0.04 \right]^{-1}$$

$$= \underline{5.71 w/m^2k}$$

The heat transmission of the frame

$$K= \left[\frac{1}{\alpha i} + \frac{s}{\lambda} + \frac{1}{\alpha o}\right]^{-1} = \left[0.13 + \frac{0.068}{0.13} + 0.04 \right]^{-1}$$

$$= \underline{1.44 w/m^2k}$$

The average of the window (glass + frame)

$$K \text{ average} = 5.71 \times 0.8 + 1.44 \times 0.2 = \underline{4.86 \text{ w/m}^2\text{k}}$$

The heat transmission through the wall with the thickness of 36.5cm

$$K = \left[\frac{1}{\alpha i} + \sum\frac{s}{\lambda} + \frac{1}{\alpha o}\right]^{-1} = \left[0.13 + \frac{0.015}{0.7} + \frac{0.365}{0.72} + \frac{0.02}{0.87} + 0.04\right]^{-1}$$

$K = 1.39$ w/m²k

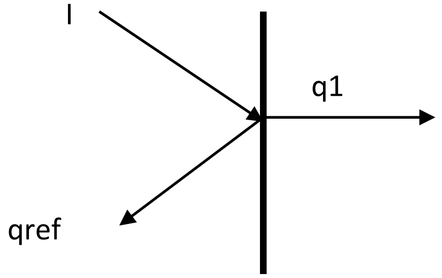
$q = 1.39\ (20\text{-}0) = 27.8$ w/m²

<u>The energy balance</u>

$$\sum qin \quad = \quad \sum qout$$

$\sum q1 + a.I \ = \ \sum qref$

<u>$a \ = 1 - \rho = 0.5$</u>



$q1 = \dfrac{\delta\alpha i - \delta I o}{\frac{1}{\alpha i} + \sum\frac{si}{\alpha i}}$ [w/m²]

$\sum\frac{si}{\alpha i} = \frac{0.015}{0.7} + \frac{0.365}{0.72} + \frac{0.02}{0.87} = 0.55$ m²k/w

$qref = \dfrac{\delta I o - \delta\alpha o}{\frac{1}{\alpha o}}$ [w/m²]

<u>$\sum q1 + a.I - \sum qref = 0$</u>

$a.I + \dfrac{\delta\alpha i - \delta I o}{\frac{1}{\alpha i} + \sum\frac{si}{\alpha i}} - \dfrac{\delta I o - \delta\alpha o}{\frac{1}{\alpha o}} = 0$

$$a.I . [\frac{1}{\alpha i} + \sum\frac{si}{\lambda i}] . \frac{1}{\alpha o} + (\delta ai - \delta lo) . \frac{1}{\alpha o} - (\delta lo - \delta ao). [\frac{1}{\alpha i} + \sum\frac{si}{\lambda i}] = 0$$

$$\delta lo = \frac{a.I.\left[\frac{1}{\alpha i} + \sum\frac{si}{\lambda i}\right].\frac{1}{\alpha o} + (\delta ai).\frac{1}{\alpha o} - (\delta ao).[\frac{1}{\alpha i} + \sum\frac{si}{\lambda i}]}{\left[\frac{1}{\alpha i} + \sum\frac{si}{\lambda i} + \frac{1}{\alpha o}\right]}$$

$$\delta lo = \frac{0.5x250 \text{ x } (0.13+0.55) \; x \; 0.04+20 \; x \; 0.04+5 \; x \; (0.13+0.55)}{0.72} = 10.56 \text{ °c}$$

$$q1 = \frac{\delta ai - \delta lo}{\frac{1}{\alpha i} + \sum\frac{si}{\alpha i}} = \frac{20-10.56}{0.13+0.55} = 13.9 \text{ w/m}^2$$

<u>The artificial heating</u>

∑q artificial heating = ∑q outside

∑q artificial heating = (Kwall .Awall +Kwindow .Awindow + nl . ρl.c .v) . (δai-δao)

= (1.39x6 + 4.86x4 + 0.5x0.35x40)(20-0)

= <u>695.6 w</u>

A wall = 0.6 x 10m² = 6m²

A window = 0.4 x 10m² = 4m²

V = 4 x 4 x 2.5 = $40m^3$

∑q = q sun + q artificial heating = I . τ. A + q artificial heating

= 250 x 0.8 x 3.2 + q artificial heating = 640 w + q artificial heating

Q w = $\frac{\delta ai - \delta lo}{\frac{1}{\alpha i} + \sum \frac{si}{\alpha i}}$. A [w] = $\frac{20-10.56}{0.13+0.55}$ x 6 = 83.3 w

<u>The transmission through the glass area</u>

Qglass= K glass. A glass (δai-δao) = 5.71 x 3.2 x (20-0) = 274.1 w

Qair = nl .ρl. c . V (δai-δao) = 0.5 x 0.35 x 40 x (20-5) = 105 w

Q frame = $\frac{\delta ai - \delta lo\ frame}{\frac{1}{\alpha i} + \frac{S\ frame}{\lambda\ frame}}$. A frame

$$\delta lo\ frame = \frac{a.I\left(\frac{1}{\alpha i}+\frac{S\,f}{\lambda\,f}\right)\frac{1}{\alpha o} + \delta ai\frac{1}{\alpha o} + \delta ao\left(\frac{1}{\alpha i}+\frac{S\,f}{\lambda\,f}\right)}{\left(\frac{1}{\alpha i}+\frac{S\,f}{\lambda\,f}+\frac{1}{\alpha o}\right)}$$

$$= \frac{0.5x250\ (0.13+0.52)\,0.04+20x0.04+5(0.13+0.52)}{0.69} = 10.58°c$$

Q frame = $\frac{20-10.58}{0.13+0.52}$ x 0.8 = 11.59 w

Q artificial heating = Qwall+Qglass+Qframe+Qair-Qsun

= 83.3+274.1+11.59+105-640 = -166.0 w

It means we don’t need artificial heating. We need to redact the warmness in the space 166 w.

تبريد الفراغ الداخلي

بهذه الطريقة التي وردت في المثال السابق يمكننا معرفة كمية الحرارة التي ينبغي سحبها للوصول إلى درجة حرارة مناسبة للفراغ الداخلي. وبالتالي فليس الهدف من هذه الأمثلة الحفاظ على الحرارة الداخلية لفراغ ما بل أيضا إدراك الكمية التي ينبغي طرحها حتي يكون هذا الفراغ ملائم لعيش الأدمي.

ولقد نوهت في العديد من كتبي على رأسها كتاب فقراء العمارة إلى الطرق غير التقليدية للمعالجات الحرارية ، منها مشروع Expo92 للمعماري نيكولاس جريم شو . ونحن في هذا المقام ندعو إلى الإقبال على تلك الممارسات خاصة إذا كان الإنسان بفضل المادة العلمية التي يقدمها هذا الكتاب المتواضع يستطيع أن يحدد:

-المادةالمستخدمة بناء على معامل التوصيل

-كمية الحرارة التي ينبغي طرحها للوصول إلى درجة حرارة ما

وعليه فليس فيزياء المباني إلا أداة تصميمية لكل من رغب في إيجاد فراغ جيد يتناسب مع تطلعات الإنسان الحرارية ويوفر في الإستهلاك الكهربي في آن معا.

Problem 13

The walls of one space are homogenous (one layer walls). The Temperature of the air inside the space is 20°c. One of the four walls has a window. The following data is for your calculation.

Given

Room Dimension: 5 x 5 x 2.5 , Windowsarea $5m^2$

Transmission intensity $3.0W/m^2k$

Short waves transmission grad 0.8

Short waves absorption grad 0

Thickness of the walls 22cm, Humidity 476 kg/m^3

Conductivity 0.4 W/mk

Specific transmission intensity 0.28 Wh/kgh

Heat absorption ability 0.35 Wh/m^3 k

Temperature inside	20°c
Air changing number	0.5 h^{-1}
Heat resistance inside	0.13 m^2k/w
Heat resistance outside	0.08 m^2k/w

Calculate

-Which orientation has the elevation? Justify your answer?

-Calculate the artificial heating and cooling in the period from 7till 12 o'clock

Solution

The south orientation is the best orientation because of energy.

To calculate the heat and cool energy we need the energy balance

∑Q out = ∑Q in

∑Q out = Q sun + Q artificial heating

∑Q in = Q wall + Q window + Q air

Q window = K. Awindow (δai –δao)

Q air = nl. 0.35 .V(δai –δao)

With

V = 5 x 5 x 2.5 = 62.5 m^3

Q sun = A window . I . τ = 5 .I . 0,8 = 4I

Q window + Q air = (15+10.94) (δai –δao) = 25.94(20-δao) [w]

Qwall = Aw $\frac{\delta ai-\delta li}{\frac{1}{\alpha i}}$= 7.5 $\frac{20-\delta li}{\frac{1}{\alpha i}}$

Aw = A –A window = 5 x 2.5 -5 =7.5 m²

<u>To calculate the artificial heating and cooling we must use Binder-Schmidt Method</u>

$$Si = \lambda \, . \, \frac{1}{\alpha i} = 0.4m \times 0.13 = 0.05m$$

$$Sa = \lambda \, . \, \frac{1}{\alpha o} = 0.4m \times 0.08 = 0.032m$$

$$\Delta x = \frac{s}{n} = \frac{0.22}{4} m = 0.055m = 5.5cm$$

$\frac{\Delta x}{2}$ = 0.0275m < si und sa

$$\Delta t = \frac{(0.055)^2}{2 \, x \, 0.4} \times 476 \times 0.28 = 0.5h$$

The calculation of the absorb temperature of any building element

$$\boxed{\boldsymbol{\delta ab = \delta ao + \frac{1}{\alpha o}(as.I)\text{-}k}}$$

δao Temperature outside

as absorption grad

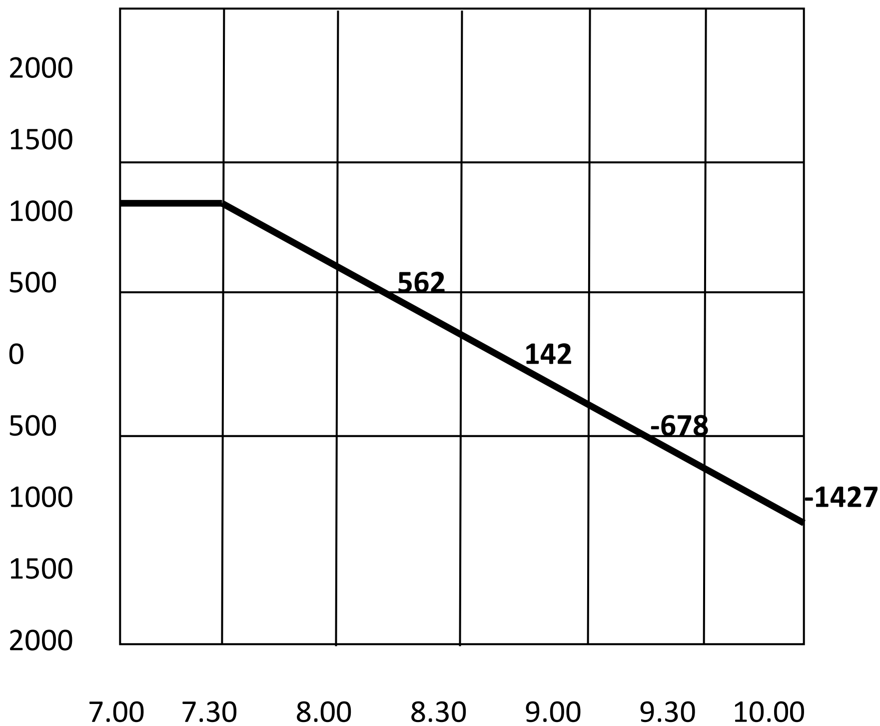
I sun intensity

K corrector factor

qa heat transmission from the sun to the outside wall .

$$qa = \frac{1}{\alpha o} (\delta ab - \delta lo) \quad [w/m^2]$$

Time	7.00	7.30	8.00	8.30	9.00	9.30	10.00
QH	562	562	142	-278	-678	-1064	-1427



Example 14

The window of one office includes frame and glass. We have to neglect the frame in the calculation. The following data is to use in the calculation.

The glass of the window ……… Thickness 4.0mm

Conductivity 0.8 W/mk

Temperature ……………… inside 22°c

……………… outside 5°c

Calculate

- The lost energy through the window?
- If we cover the glass in a distance of 1cm with a material 0,06 w/mk and 5mm thickness, how much is the lost energy
- If the resistance of the cover 0.5m²k/w and the conductivity of the air layer 0.02w/mk please calculate
  - **A)** the reduction of the heat transmission
  - **B)** the radiation between the two glass level
  - **C)** the lost energy in the air layer.

Solution

The Heat transmission through the glass

q glass = kg .(δai-δao)

K glass = $[\frac{1}{\alpha i}+\frac{s}{\lambda}+\frac{1}{\alpha o}]^{-1}$ = $[\,0.13+\frac{0.004}{0.8}+0.04]^{-1}$= 5.71 w/m²k

q glass = 5.71 (22-5) = 97.1w/m²

$\eta = \frac{qG-q}{qG}$. 100 [%]

q = k. (δai- δao) = 2.51 (22-5) = 42.7w/m²

$K = \left(\frac{1}{\alpha i}+\frac{s}{\lambda}+\frac{1}{\Lambda air}+\frac{S\,glass}{\lambda\,glass}+\frac{1}{\alpha o}\right)^{-1}$

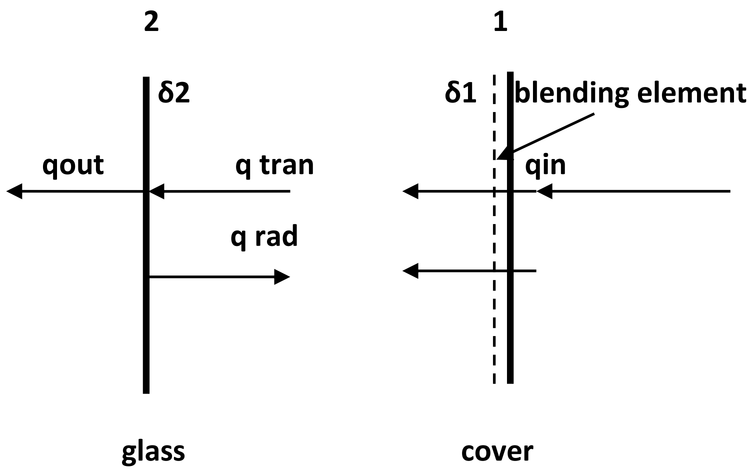
$K = \left(0.13+\frac{0.005}{0.06}+0.14+\frac{0.004}{0.8}+0.04\right)^{-1}$

= 2.51 w/m²k

$\eta = \frac{97.1-42.7}{97.1}$. 100 = 56%

3a) the reduction of the heat transmission

$\eta 1 = \frac{qG-g'}{qG}$. 100 ; $\eta 2 = \frac{q-g'}{q}$. 100



we calculate it through heat balance

q in = q rad + q tran = q' = $\frac{\delta ai - \delta 1}{\frac{1}{\alpha i} + \frac{1}{\Lambda}}$ [w/m²]

q out = q rad + q tran = q' = $\frac{\delta ai - \delta 1}{\frac{1}{\Lambda} + \frac{1}{\alpha o}}$ [w/m²]

<u>in the air layer</u>

q' =q rad + q tran [w/m²]

q rad = $\frac{\delta 1 - \delta 2}{\frac{1}{\alpha\, rad}}$ [w/m²]

q tran = $\frac{\delta 1 - \delta 2}{\frac{1}{\Lambda\, air}}$[w/m²]

but ....

$\delta 1 = \delta ai - q\left(\frac{1}{\alpha i} + \frac{1}{\Lambda}\right)$

$\delta 2 = \delta ao - q\left(\frac{1}{\Lambda} + \frac{1}{\alpha o}\right)$

q rad = $\frac{\delta 1 - \delta 2}{\frac{1}{\alpha\, rad}} = \frac{\delta ai - q'\left(\frac{1}{\alpha i} + \frac{1}{\Lambda cover}\right) - \delta ao - q'\left(\frac{1}{\Lambda\, glass} + \frac{1}{\alpha o}\right)}{\frac{1}{\alpha\, rad}}$

$$= \frac{(\delta ai - \delta ao) - q'\left(\frac{1}{\alpha i} + \frac{1}{\Lambda cover} + \frac{1}{\Lambda\, glass} + \frac{1}{\alpha o}\right)}{\frac{1}{\alpha\, rad}} \quad [w/m^2]$$

$$q\ tran = \frac{\delta 1 - \delta 2}{\frac{1}{\Lambda\, air}} = \frac{\delta ai - q'\left(\frac{1}{\alpha i} + \frac{1}{\Lambda cover}\right) - \delta ao - q'\left(\frac{1}{\Lambda\, glass} + \frac{1}{\alpha o}\right)}{\frac{1}{\Lambda\, air}}$$

$$= \frac{(\delta ai - \delta ao) - q'\left(\frac{1}{\alpha i} + \frac{1}{\Lambda cover} + \frac{1}{\Lambda\, glass} + \frac{1}{\alpha o}\right)}{\frac{1}{\Lambda\, air}}$$

q' =q rad + q tran

$$q' = \frac{(\delta ai - \delta ao)\left(\frac{1}{\alpha\, rad} + \frac{1}{\Lambda\, air}\right)}{\left(\frac{1}{\alpha\, rad} + \frac{1}{\Lambda\, air}\right) + \left(\frac{1}{\alpha\, rad} + \frac{1}{\Lambda\, air}\right)\left(\frac{1}{\alpha i} + \frac{1}{\Lambda cover} + \frac{1}{\Lambda\, glass} + \frac{1}{\alpha o}\right)}$$

$$q' = \frac{(22-5)\left(\frac{1}{2} + \frac{0.01}{0.02}\right)}{\frac{0.01}{0.02} \cdot \frac{1}{2} + \left(\frac{1}{2} + \frac{0.01}{0.02}\right)\left(0.13 + \frac{0.005}{0.06} + \frac{0.004}{0.8} + 0.04\right)} = 33.4\ w/m^2$$

$$\eta 1 = \frac{97.1 - 33.4}{97.1} .\ 100 = 66\%$$

$$\eta 2 = \frac{42.7 - 33.4}{42.7} .\ 100 = 22\%$$

The radiation coefficient

$$C1,2 = \frac{q\ rad}{\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4}$$

$q\ rad\ = (\frac{1}{\alpha\ rad})^{-1}.\ (\delta1 - \delta2)$

$T1 = \delta1 + 273 = 14.9 + 273 = 287.9\ k$

$T2 = \delta2 + 273 = 6.5 + 273 = 179.5\ k$

$\delta1 = \delta ai - q(\frac{1}{\alpha i} + \frac{1}{\Lambda}) = 22 - 33.4(0.13 + \frac{0.005}{0.06}) = 14.9°c$

$\delta2 = \delta ai - q(\frac{1}{\Lambda} + \frac{1}{\alpha o}) = 5 + 33.4(\frac{0.004}{0.8} + 0.04) = 6.5°c$

$q\ rad\ = (\frac{1}{\alpha\ rad})^{-1}.\ (\delta1 - \delta2) = \frac{14.9 - 6.5}{0.5} = 16.8 w/m^2$

$C1,2 = \frac{16.8}{(\frac{287.9}{100})^4 - (\frac{279.5}{100})^4} = 2.2\ w/m^2 k^4$

$q\ tran = q - q\ rad = 33.4 - 16.8 = 16.6$

$\eta\ tran = \frac{33.4 - 16.8}{33.4} = 0.5 = 50\%$

Problem 15

In a room with an Air-condition the constant temperature is 18°. The air changing number is $0.1h^{-1}$. The sun intensity is $400 w/m^2$

Roomdimension 6 x 4 x 2.5m

Window	$2m^2$
Frame	20%

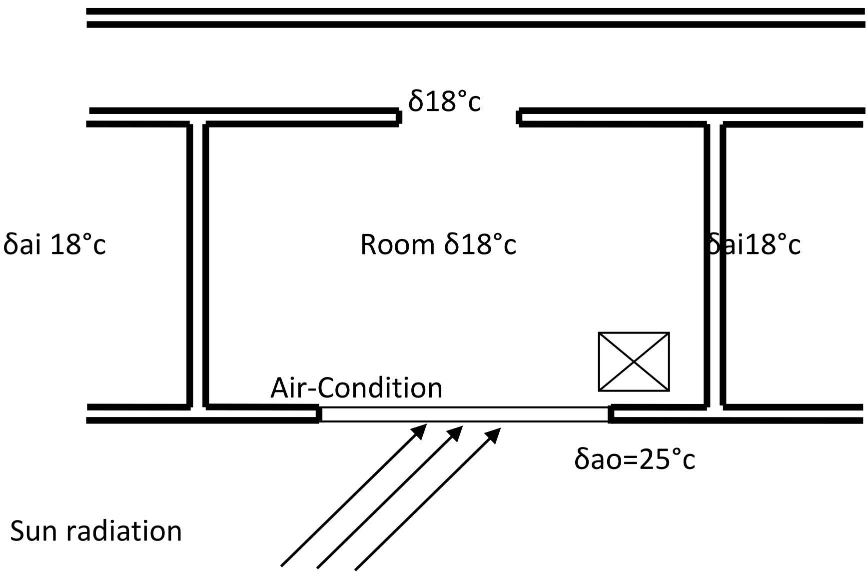
Thermal Conductivity 2.6w/m^2k

Transmission grad (glass) 0.8

Outside wall 36.5cm λ = 0.6 w/mk

Absorption grad of the level outside (only short waves) a=0.4

Heat capacity of the air 0.35 wh/m^3k



Calculate

- -The Heat transmission through the outside wall
- -The temperature outside and inside of the outside wall (elevation)
- -The cooling energy

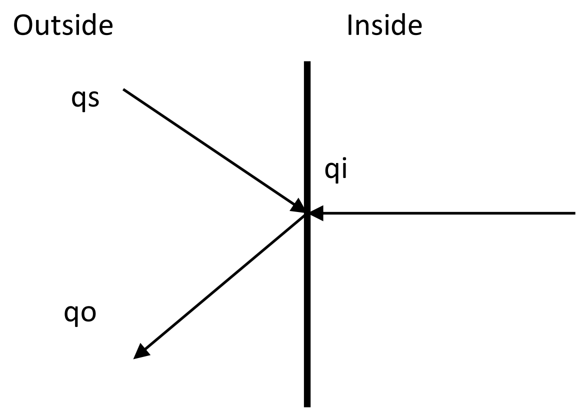
Solution

The Heat transmission coefficient, U-Value of the wall

K = $\left(\frac{1}{\alpha i}+\frac{s}{\lambda}+\frac{1}{\alpha o}\right)^{-1}$ = $\left(0.13+\frac{0.365}{0.6}+0.04\right)^{-1}$ = 1.28 w/m²k

<u>To calculate the outside temperature we must use the energy balance</u>

$$\sum q = qs - qi - qo = 0$$



q_s = I.a = 400 x 0.4 = 160 w/m²

qi = $\frac{\delta lo-\delta ai}{\frac{1}{\alpha i}+\frac{s}{\lambda}}$

qo = $\frac{\delta lo-\delta ao}{\frac{1}{\alpha o}}$

$$\mathbf{I.a} - \frac{\delta lo-\delta ai}{\frac{1}{\alpha i}+\frac{s}{\lambda}} - \frac{\delta lo-\delta ao}{\frac{1}{\alpha o}} = 0$$

$\delta lo = k \cdot [\, I.a \cdot \frac{1}{\alpha o} (\frac{1}{\alpha i} + \frac{s}{\lambda}) + \delta ai \cdot \frac{1}{\alpha o} + \delta ao \cdot (\frac{1}{\alpha i} + \frac{s}{\lambda})] = 30.6°c$

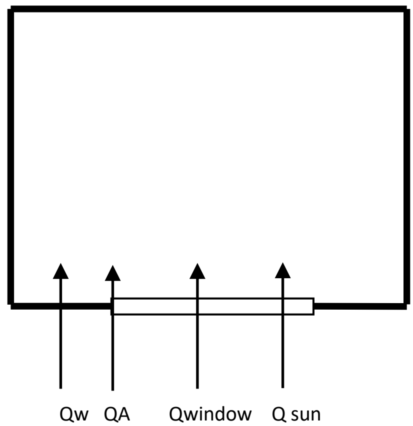
$q_o = \frac{30.6-25}{0.04} = \underline{140\ w/m^2}$

$q_i = \frac{\delta lo - \delta ai}{\frac{1}{\alpha i}} = qs - qo = 160 - 140 = 20\ w/m^2$

$\delta li = 20 \times 0.13 + \delta ai = 2.6 + 18 = 20.6°c$

3-The cooling energy

To calculate the cooling energy we must use the heat balance.



$\sum Q = Qwall + Qair + Q\ window + Q\ sun - Q\ cooling = 0$

$Qwall = q.Aw = 20 \times 13 = \underline{260w}$

$A_w = A - A_{window} = 6x2.5 - 2 = 13m^2$

$Q_{window} = k_w . A_w(\delta_{ao}-\delta_{ai}) = 2.6 \times 2\ (25-18) = 36.4w$

$Q_{sun} = I.\tau.A_{glass} = 400 \times 0.8 \times 1.6 = 512w$

$A_{glass} = A_{window} - 0.2\ A_{window} = 0.8\ A_{window} = 1.6m^2$

$Q_{air} = n . \rho . c . v\ (\delta_{ai}-\delta_{ao}) = 0.1 \times 60 \times 0.35 \times (25-18) = 14.7\ w$

$V = 6 \times 4 \times 2.5 = 60m^3$

<u>The cooling energy for the space</u>

$Q_{cooling} = 260 + 36.4 + 512 + 14.7 = \underline{823.1\ w}$

<u>Problem 16</u>

The wall outside has the thickness of 15cm (homogenous wall) with the following data :

Density 2000kg/m³

Conductivity 1.4 w/mk

Specific heat capacity 0.28wh/kgh

Time	8.00	8.15	8.30	8.45	9.00	9.15	9.30	9.45	10.00
δai	20	21	22	23	24	24	24	24	24
δao	6	7	9	10	12	14	15	16	18

Calculate

-The temperature in the cross section from 8.00 till 10.00.

-The temperature on the level inside in the same period.

Solution

Si = $\lambda \cdot \frac{1}{\alpha i}$ = 1.4 x 0.13 = 18.2cm

Sa = $\lambda \cdot \frac{1}{\alpha a}$ = 1.4 x 0.04 = 5.6cm

Layers Thickness

$$\Delta x = \frac{s}{n} = \frac{0.15}{3} = 0.05m = 5cm$$

Stability

$\frac{\Delta x}{2}$ **= 2.5cm < si and sa**

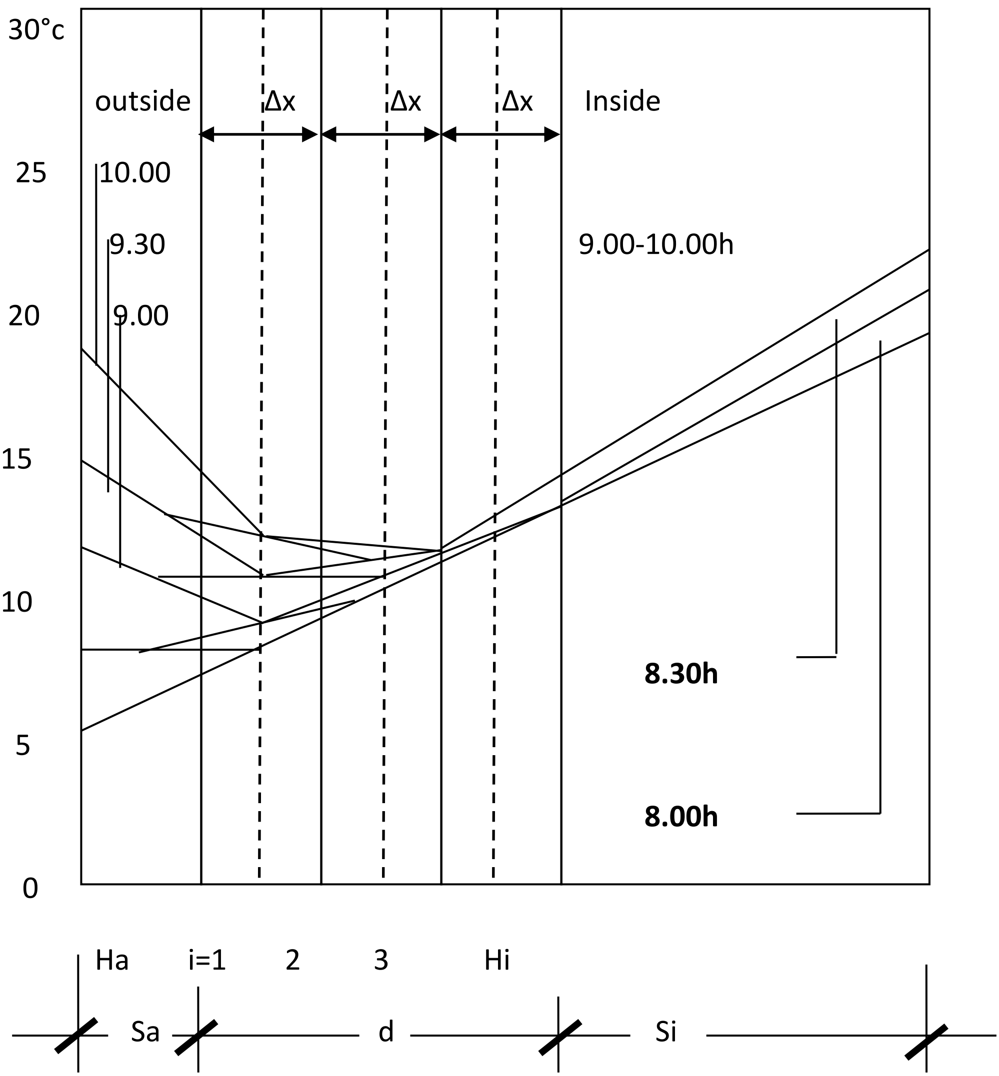
Time

$$\Delta t = \frac{(\Delta x)^2}{2.a} [h]$$

$$a = \frac{\lambda}{\rho.c} = \frac{1.4}{2000 \; x \; 0.28} = 2.5 \cdot 10^{-3} \; m^2/h$$

$$\Delta t = \frac{(0.05)^2}{2 .2,5} 10^3 = 0.5 \; h$$

Time	8.00	8.30	9.00	9.30	10.00
Δli	13.4	13.7	14.2	14.6	15.0
Δlo	8.1	9.0	10.7	12.2	14.2



أستغرب كثيرا من مهندسي قسم ميكانيكا ، الذين يفضلون استخدام برامج المحاكاة وفي مقدورهم حساب الطاقة اللازمة للتكيف بأبسط الطرق وعلى مدار ساعتين متصلتين في المثال السابق:

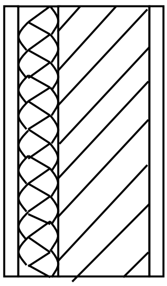
أذكر أثناء دراستي بجامعة شتوتجارت أنني دراسة على يد Prof.Bach

أستاذ الطاقة بقسم الهندسة الميكانيكية ، وكان علي أن أتعلم برنامج Transys ، وبالمناسبة البرنامج لايزال متداول في الأسواق ، وعلى الرغم من إجادتي لهذا البرنامج إلا أنه كان يصر على أن أقوم بحساب الكثير من العمليات المعقدة على يدي ، ولازلت أذكر حساباتي لل Turblance flow - Fuid Mechanics .

لكن الجيل الحالي من المهندسين ويظهر هذا في رسائل الماجستير لايتردد في استخدام تلك البرامج دون إجادة للخلفية الفيزيائية التي قامت عليها. هذا للأسف

Problem 17

In an office they are sitting 2 people. The façade of the room include 30% isolated glass. The sun radiation is 600 W/m² and the time for this radiation is 8 hours.



2.0cm Cement λ=0.87 w/mk

4.0cm isolation λ= 0.04 w/mk

24.0cm masonry λ=0.36 w/mk

1.5cm Cement λ=0.70 w/mk

Elevation Area 12.5m², b= 4m

Glass heat transmission coefficient 3.1w/m²k

Transmission grad 0.8

Person heat 120w

Transmission grad of the wall 0.6

Air capacity 0.35 wh/m³k, Air temperature 6°c

Calculate

- The thermal transmission coefficient of the wall
- The radiation energy from the window in the space
- If the air temperature of the space inside 20°c constant

  Please calculate:

  The lost energy through the wall

  The lost energy through the window

  How much is the air change between inside and outside to have constant temperature of 20°c in the space inside.

Solution

-The thermal conductivity of the wall

$$K = \left(0.13 + \frac{0.015}{0.7} + \frac{0.24}{0.36} + \frac{0.04}{0.04} + \frac{0.02}{0.87} + 0.04 \right)^{-1} = 0.53 \text{ w/m}^2\text{k}$$

<u>-The radiation energy through the window</u>

Q rad = I . t .τ .A = 600 x 8 x 0.8 x 0.3 x 12.5 = 14.4 kw/h

-Transmission heat through the wall

Qw = k.(δai- δs).A

A =12.5 (1-0.3) = 8.75m²

The outside temperature δs

$\delta s = \delta ao + \frac{1}{\alpha o} . a . I = 6 + 0.04 \times 0.6 \times 600 = 20.4°c$

$Qw = 0.53 (20\text{-}20.4) \times 8.75 = -1.9w$

<u>-Transmission heat through the window</u>

$-Qwindow = k.(\delta ai - \delta ao).A = 3.1 (20\text{-}6) \times 3.75 = 162.75w$

$A = 12.5 \times 0.3 = 3.75m^2$

<u>To calculate the Air-changing number</u>

$\sum Q$ **= Q rad + Q p – Qwall + Qwindow –Qa = 0**

$Q\ rad = I . \tau . A = 600$

$Q\ p = 2 \times 120 = 240$

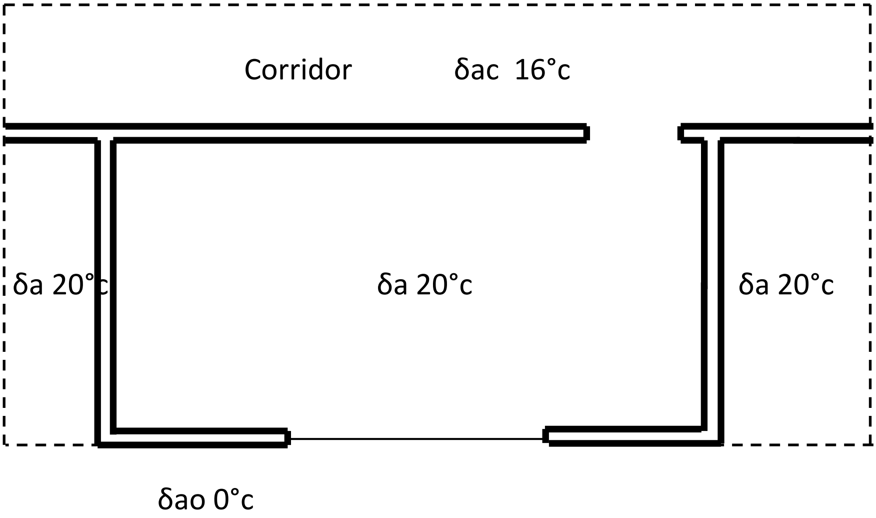
$Q\ a = \rho . c . n. V (\delta ai\text{-}\delta ao) = 0.35 \times 12.5 \times 4 \times 14 \times n$

$\sum Q = 600 \times 12.5 \times 0.3 \times 0.8 + 240 + 1.9 - 162.75 – 245.n = 0$

$n = \frac{187.15}{245} = 7.7\ h^{-1}$

<u>Problem 18</u>

In an office they are sitting 2 people. The façade of the room include 60% isolated glass. The energy inside the space is from two components lighting and human body of the two people.



Room Dimension 7.2 x 4.8 x 2.85

Door Dimension 1.0 x 2.0

Conductivity of the door 2.0 w/m^2k

Window conductivity = k w = kframe = 2.1 w/m^2k, g = 0.8

Frame 20%

Building inside wall 2.0 w/m^2k

Wall outside

1.5cm cement	λ = 0.7 w/mk
24cm masonry	λ = 0.6 w/mk
8.0cm Isolation	λ = 0.035 w/mk
0.5cm cement	λ = 0.7 w/mk

Air capacity of heat	0.28 wh/kgk
Air density	1.25kg/m³

Energy

Energy of the people	100w per person
Lighting	300W
Office machine	400W
Air changing number 50% door , 50% window	1.0 h^{-1}
Sun intensity	600W/m²

Calculate

-The thermal transmission coefficient of the wall
-The thermal balance
-Which adding energy of cooling or heating to have a constant temperature of 20°c.

Solution

-The thermal conductivity of the wall

$$K = \left(0.13 + \frac{0.015}{0.7} + \frac{0.24}{0.6} + \frac{0.08}{0.035} + 0.04 \right)^{-1} = 0.35 \text{ w/m}^2\text{k}$$

-The thermal balance

$\sum Q$ = Q in – Qout + Q heat, cool = 0

Qin = Q sun + Q per + Q bel + Q office machine =

Q sun = I . g . A = 600 x 0.8 x 9.8 = 4704 w

Q in = 4704 + 200 + 300 + 400 = 5604 w

The Area what we use

A wall = 7.2 x 2.85 x 0.6 = 12.3m²

A glass = 12.3 x 0.8 = 9.8 m²

Q out = Qw + Qwindow +Qroom + Qa +Qw.inside + Qdoor +Qa door

= (0.35 x 8.2 + 2.1(9.8+2.5) + 1.0 x 0.5x 1.25 x0.28 x 98.5)(20-0)

+ (2 x 18.5 + 2 x 2.0+ 1.0x0.5 x 1.25 x 0.28 x 98.5)(20-16)

= 1151.7 w

$\sum q$ = qin +qout + q heat, cool= 0

= 5604 – 1151,7 + q heat, cool =0

= - 4452w = -4.5w cooling energy

Moisture and diffusion

Sources of moisture

The sources of the moisture in the space or the buildings layers are as following:

-Indoor humidity (occupants, cooking, bathing, washing/ drying clothes)

-Construction moisture (typically higher in initial phase)

-Precipitation (rain, snow hail)

-Water leakage

-Liquid water and water vapor in the ground.

Water vapor presence in air

$$P_v = R_v \cdot T_{abs} \cdot \sigma$$

P_v: Partial water vapor pressure

R_v: Specific gas constant for water vapor 461,52J/kgk

σ: water vapor concentration kg/ m3

T_{abs} : Temperature [k]

T: Temperature [°c]

Maximum possible water vapor concentration in air σs [kg/m3]

$$\sigma_s = \frac{a.(b+0,01T)^n}{Rv(T+273,15)} \text{ [kg/m3]}$$

ما الفارق بين رطوبة الجو Humidityورطوبة أجزاء المبنىMoisture

الفارق هو الفارق المكاني ،فرطوبة الجو بسبب بجار الماء المتواجد في جزيئات الأثير أو الفوتونات أما الرطوبة المتواجدة داخل الجدران ، والتي تسبب أضرار للمبنى ، قد تؤدي إلى إنهياره عندما تصل إلى حديد التسليح فقد تكون بسبب بخار الماء وقد تكون بسبب المياة الجوفية المتسربة عبر الأساسات إلى الجدران.

وواجبنا نحن تحديد المكان ومعرفة كمية المياة المتواجدة. وهذا لن يكون إلا بالتعرف على نظرية Glaser . والتي سنشرحها لاحقا.

Moisture and building layers

The water mass exist in every layer of the walls. To calculate this mass we must know the pressure at the edges of the layers.

$$m1=m2 = \frac{\Delta p}{1{,}5.10^6 \sum si.\mu i}$$

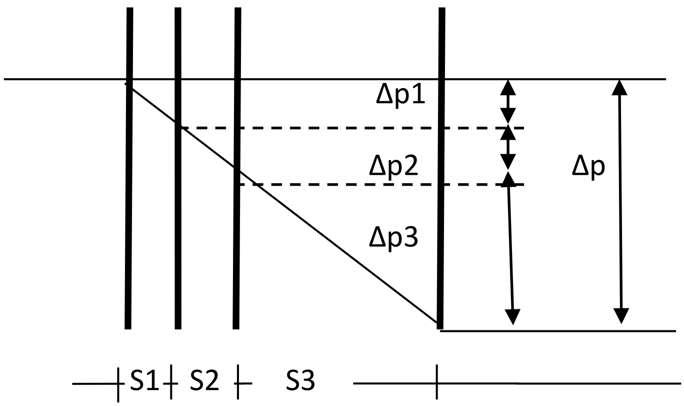


Fig.1:11 the relation between the thickness of the layer and the difussions resistance

Which

S = the thickness of the layer

μ= the diffusion resistance

The diffusion flow is equal the diffusion pro Area. It means the following equation:

$$M = A \cdot m = \frac{\Delta p \cdot A}{1{,}5 \cdot 10^6 \cdot sd}$$

Humidity

The humidity is the water molecules in the gas form of the Air. The temperature of the Air is much related to the humidity.

δin °c	Cs in g/m³	δ in °c	Cs in g/m³
-15	1.39	15	12.8
-5	3.23	20	17.3
0	4.85	25	23.0
5	6.79	27	25.8
10	9.39	29	28.8

The relative humidity is the dividing from the water mass in the air to the maximum water mass from this temperature.

$$\Phi = \frac{c}{cs} = \frac{p}{ps}$$

δ °c	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8
30	4244	4269	4294	4319	4344	4369	4394	4419	4445
20	2340	2354	2369	2384	2399	2413	2428	2443	2457
10	1228	1237	1245	1254	1262	1270	1279	1287	1296
0	611	616	621	626	630	635	640	645	648
-5	401	398	395	391	388	385	382	379	375
-10	260	258	255	253	251	249	246	244	242
-15	165	164	162	161	159	158	157	155	153

$$P = c \, . \, R \, . \, T$$

$$Ps = a \left(b + \frac{\delta}{100} \right)^{n}$$

	$0 \le \delta \le 30$	$-20 \le \delta \le 0$
A	288.6 Pa	4.68 Pa
B	1.09	1.48
N	8.02	12.30

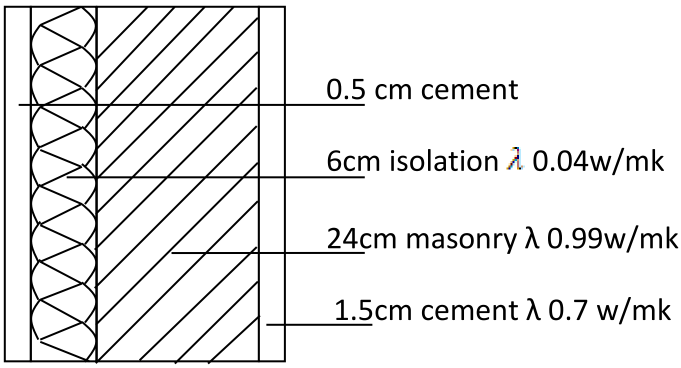
نظرية Glaser

تعتمد نظرية على رسم منحنى للضغط عبر قطاع الحائط ومنحنى آخر للضغط المشبع أو المحمل بجزيئات الماء ، فإذا تقاطع المنحنيان يكون في هذا المكان من الجدار ترسيب مائي ، وإذا لم يتقاطعا لا يوجد ترسيب مائي.

مع الأخذ في الإعتبار أننا حتى نحصل على معدلات الضغط في أماكن متفرقة داخل قطاع الحائط فلابد من معرفة درجات الحرارة أولا.

Problem 19

The façade of an office building includes isolated wall and glass part. The design of the wall is like the shown diagram. Please test the appearance of condense water on the level inside.



Window 12mm Air layer $1/\Delta = 0.14\ m^2k/w$

Air temperature inside 20°c

Outside -15°c

Relative humidity 50%

Thermal resistance inside $0.17m^2k/w$

Thermal resistance outside $0.04m^2k/w$

<u>Calculate</u>

-Draw the thermal diagram through the wall and the window
-Chick the condense water(Moisture) in the wall
-Which thickness of the isolation to avoid the condense water is requird

<u>Solution</u>

$$q = k.\ (\delta ai - \delta ao) \quad [w/m^2]$$

$$k\ wall = \left(0.17 + \frac{0.015}{0.7} + \frac{0.24}{0.99} + \frac{0.06}{0.04} + 0.04 \right)^{-1} = 0.51 w/m^2k$$

$$k\ glass = (0.17 + 0.14 + 0.04)^{-1} = 2.86\ w/m^2k$$

$$q\ wall = kw\ (\delta ai - \delta ao) = 0.51\ (20 + 15) = 17.9 w/m^2$$

$$q\ glass = kg\ (\delta ai - \delta ao) = 2.86\ (20 + 15) = 100.1\ w/m$$

Wall-Temperature

$\delta li = \delta ai - q\frac{1}{\alpha i} = 17.0°c$

$\delta 1 = \delta ai - q\left(\frac{1}{\alpha i} + \frac{0.015}{0.7}\right) = 16.6°c$

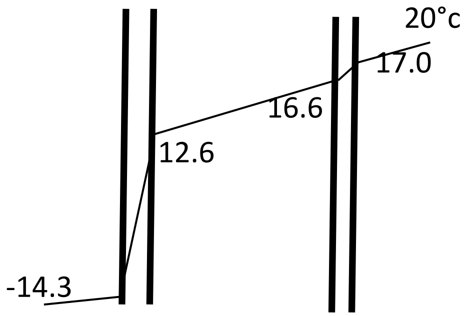
$\delta 2 = \delta ai - q\left(\frac{1}{\alpha i} + \frac{0.06}{0.04}\right) = 12.6°c$

$\delta lo = \delta ao + q\frac{1}{\alpha o} = -14.3°c$

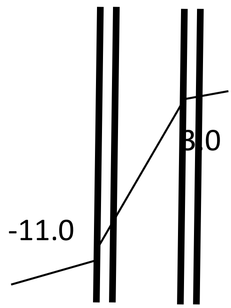
Glass-Temperature

$\delta li = \delta ai - qg\frac{1}{\alpha i} = 3.0°c$

$\delta lo = \delta ao + q\frac{1}{\alpha o} = -11.0\ °c$



WALL



GLASS

The conditions to avoid condense water on the level outside Psoi ≤Pai

P = ϕ . Ps = 0.5 x 2342 = 1171 Pa

Ps = 1941 Pa > P no condense water

On the glass inside Ps = 758 < P. We have condense water on the glass inside.

3) The condition for no condensing on the wall inside is that δli >δs

Which is δs the temperature by the saturated pressure.

Through the thermal balance equation we ll get the thickness of the Isolation.

qi = qo

$$(\delta ai - \delta li)\left(\frac{1}{\alpha i}\right)^{-1} = (\delta li - \delta ao)\left(\frac{0.015}{0.7} + \frac{0.24}{0.99} + \frac{S}{0.04} + \frac{1}{\alpha o}\right)^{-1}$$

$$S = \left[\frac{1}{\alpha i} \cdot \frac{\delta li - \delta ao}{\delta ai - \delta li} - \left(\frac{0.015}{0.7} + \frac{0.24}{0.99} + \frac{1}{\alpha o}\right)\right] . 0{,}04$$

$$= \left[\frac{0.17 \ x\ (9.3+15)}{20-9.3} - \left(\frac{0.015}{0.7} + \frac{0.24}{0.99} + 0.04\right)\right] . 0{,}04 = 3{,}3 \ . \ 10^{-3}$$

The thickness of the isolation 3.3 mm

يلاحظ في منحنى Glaser أن مانكتبه في العداد الأفقي ليس سمك الطبقات المختلفة المكونة للجدلر ، ولكنه السمك مضروبا في ثابت إنتشار الرطوبة . وهو الذي نعبر عنه ب **S . µ**

قد يختلف معامل إنتشار من مادة إلى أخرى ، يترتب على هذا حساب كمية المياة المترسبة داخل الجدار بالقانون التالي

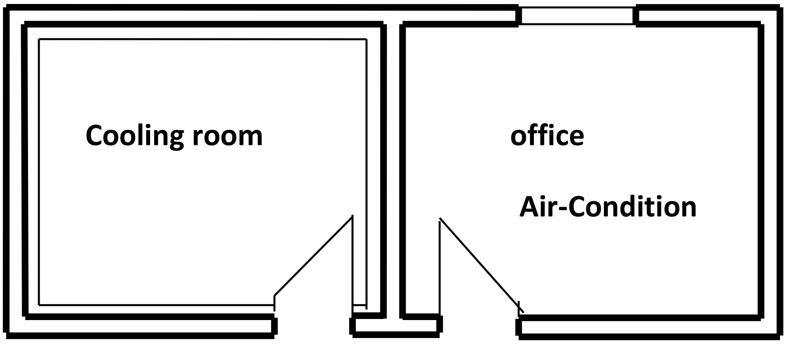
$$\mathbf{m} = \frac{p1-p2}{1/\Delta}$$

$$1/\Delta = 1.48 \times 10^{6} \sum \mu.s$$

يترتب على هذا إتخاذ القرارات التصـميمية لمعالجـة الترسـيب المـائي (الرطوبة) كل حالة على حدة .

Problem 20

In a factory we have a typical case like the following sketch. The cooling room is back to back to the office room.



Room dimension 9 x 4 x 3.5

Outside wall 24 cm λ= 0.16w/mk μ = 5

Inside wall 20cm λ= 0.21w/mk μ = 5

Isolation material resistance 2.0 m^2k/w μ s =20m

Air-temperature	outside 20°c
	outside 20°c
	Cooling 5°c
	Hall 17°c

Relative humidity	outside 50%
	Office 40%
	Cooling 80%
	Hall 60%

Calculate

-By which relative humidity are we condense water on the surface of the wall

-The mass of the condense water on the wall

-The mass balance of condense water

Solution

$\delta li = \delta lo - q \frac{1}{\alpha i}$

q= k. (δl office – δl cooling) = 0.31 x 15 = 4.7 w/m²

with

$k = \left(0.17 + \frac{0.2}{0.21} + 2 + 0.13\right)^{-1} = 0.31 w/m^2k$

The temperature of the surface of the wall

$\delta li = 20 - 4.7 \times 0.17 = 19.2°c$

The saturated pressure related to the temperature

$Ps(20°c) = 2342$ Pa

Ps(19.2°c) = 2228 Pa

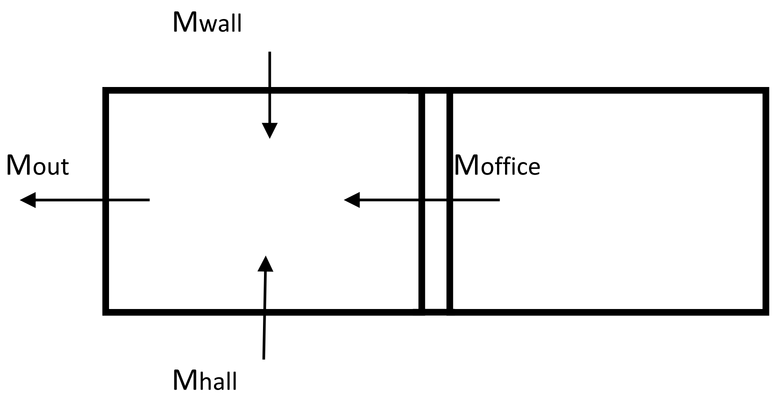
The normal pressure p= ϕ . ps

P office = 0.4 x 2342 = 937

The relative humidity

$\phi = \frac{p}{ps} = \frac{937}{2228}$ = 0.42 = 42%

The mass balance per hour



Mout = Mwall +Moffice +Mhall

M = m . A

m = $\frac{p1-p2}{1/\Delta}$

1/Δ = 1.48 x 10^6 ∑μ.s

M wall = [1.48x10^6(0.24x5 +20)$]^{-1}$.(0.5 x 2342 – 08 x 873) 45.5

= 6.85.10^{-4} kg/h

M office = $[1.48 x 10^{6} (0.2x5 +20)]^{-1}$.(0.4 x 2342 – 08 x 873) 14

= $1,07.10^{-4}$ kg/h

M hall = $[1.48 x 10^{6} (0.2x5 +20)]^{-1}$.(0.6 x 1941 – 08 x 873) 31,5

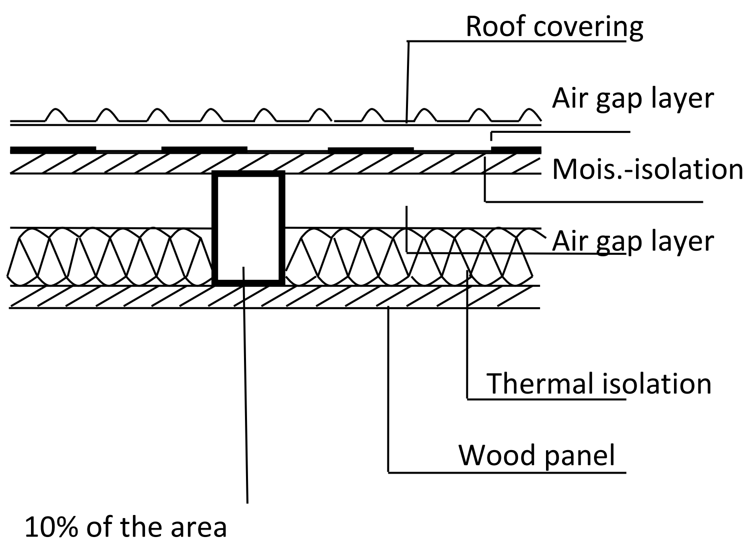
= $4,73.10^{-4}$ kg/h

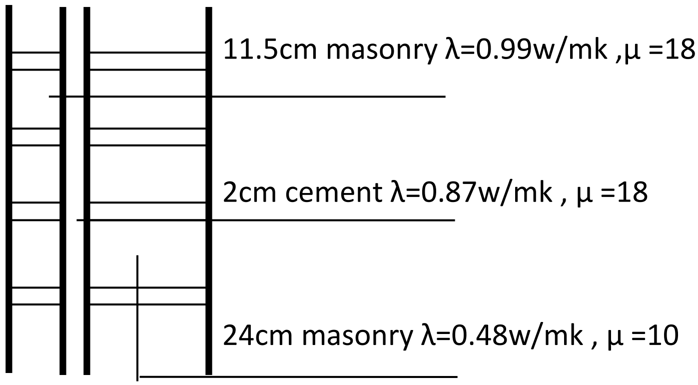
Then

Mout = 1.265 x 10^{-3} kg/h = 1.27 g/h

Problem 21

In a wood structure design as a construction solution for the roof and masonry wall for the walls please prove the condensing water availability through **glaser-methode.**





Air temperature

Winter	Inside 20°c
	Outside -10°c
Summer	Inside 12°c
	Outside 12°c

Relative humidity

Winter	Inside 50%
	Outside 80%
Summer	Inside 70%
	Outside 70%

Calculate

-Is the roof construction from the aspect of the building physics save?

-Are we having condensing in the wall section? If yes where?

-Which criteria can we us to avoid condensing?

Solution

No it is not save.

The water isolation is not in the right place. The right place is between the Thermal isolation and the wood panels

The conductivity

$K = \left(0.13 + \frac{0.115}{0.99} + \frac{0.02}{0.87} + \frac{0.24}{0.48} + 0.04 \right)^{-1} = 1.24$ w/m²k

Q = K. (δai- δao) = 1.24 (20+10) = 37.2 w/m²

The temperature and the pressure

$\delta li = \delta ai - q\left(\frac{1}{\alpha i}\right) = 20 - 37.2(0.13) = 15.2°c$ ps = 1730 pa

$\delta 1 = 20 - 37.2\left(0.13 + \frac{0.24}{0.48}\right) = -3.4°c$ 460 pa

$\delta 2 = -10 + 37.2\left(0.04 + \frac{0.115}{0.99}\right) = -4.2°c$ 430 pa

$\delta 2 = -10 + 37.2(0.04) = -8.5°c$ 297 pa

δai = δao = 12°c = constant ⟶ Ps= 1404 pa

Winter

p = ϕ . ps = 0.5 x 2342 = 1171pa

p2 = ϕ . ps = 0.8 x 260 = 208 pa

Summer

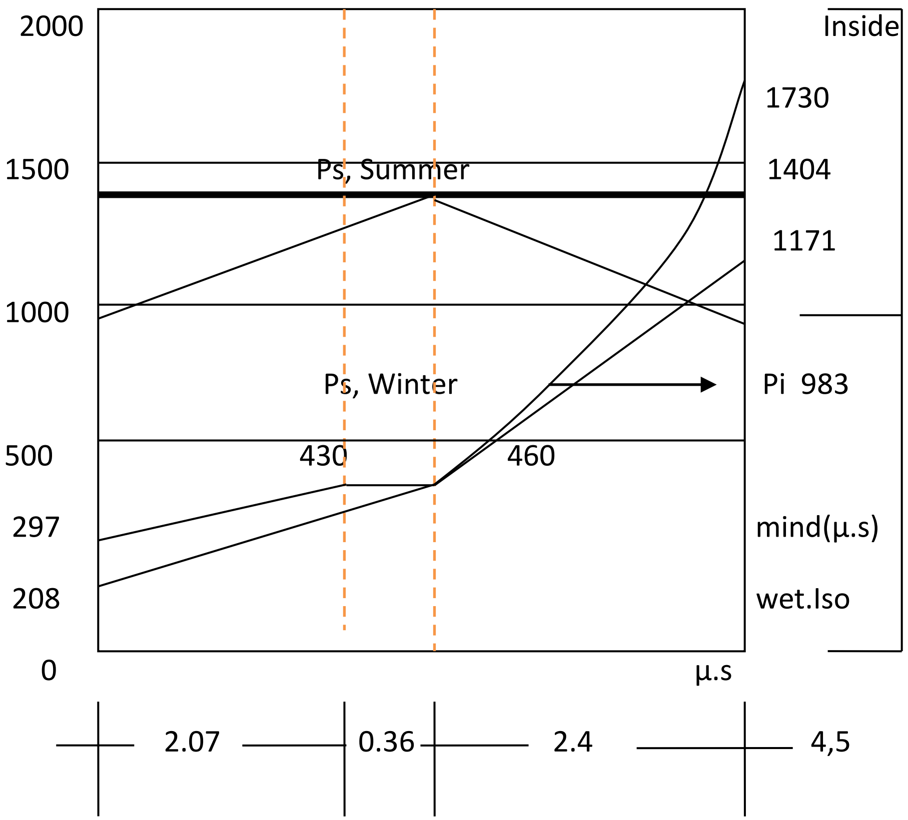
p = ϕ . ps = 0.7 x 1404 = 983 pa

Diffusion of the water vapor as layers

(μ.s)1 = 0.24 .10 = 2.4m

(μ.s)2 = 0.02 .18 = 0.36m

(μ.s)3 = 0.115 .18 = 2.07m



Mass of condense water-Moisture

$(\frac{1}{\Delta}) = 1.48.\ 10^6 \times 2.4 = 3.55\ .10^6$ m²hPa/kg

Mass of evaporation water

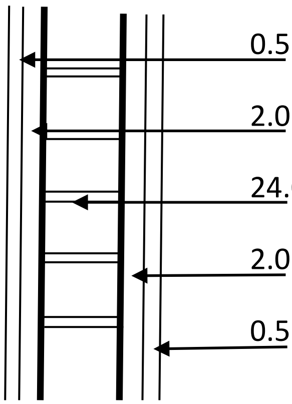
$(\frac{1}{\Delta}) = 1.48.\ 10^6 \times (0.36 + 2.07) = 3.6\ .10^6$ m²hPa/kg

$$M_{cond} = 1440\ \left(\frac{1171-460}{3.55\ .10^6} - \frac{460-208}{3.6\ .10^6}\right) = 0.188 kg/m^2 = 188 g/m^2$$

$$M_{evap} = 2160\ \left(\frac{1404-983}{3.55\ .10^6} + \frac{1404-983}{3.6\ .10^6}\right) = 0.509 kg/m^2 = 509 g/m^2$$

Problem 22

Two Apartments are placing symmetrical to the stairs. The bathrooms are back to back. In between there is one sanitary wall.



0.5 cm ceramic	λ = 1.00 w/mk	μ =200
2.0 cm cement	λ = 1.40 w/mk	μ = 15
24.0 cm masonry	λ = 0.99 w/mk	μ = 15
2.0 cm cement	λ = 1.40 w/mk	μ = 15
0.5 cm ceramic	λ = 1.00 w/mk	μ =200

Air temperature	Bath 1 24°c
	Bath 2 18°c

Humidity	Bath 1 80%
	Bath 2 80%

Calculate

-Are we have between the two bathrooms water diffusion
-Test the condensing on the wall surface of the bathroom
-How much is the relative humidity to avoid moisture

Solution

-Yes we have water diffusion because we have the same humidity and different temperature at the same time.
-to test the condensing we must use Glaser method:

$$K = \left(0.13 + \frac{0.005}{1} + \frac{0.02}{1.4} + \frac{0.24}{0.99} + \frac{0.02}{1.4} + \frac{0.005}{1} + 0.13 \right)^{-1}$$
$$= 1.85 w/m^2k$$

$Q = k (\delta a\ bath1 - \delta a\ bath2) = 1.85\ (24 - 18) = 11.1\ w/m^2$

$\delta 1 = 24 - 11{,}1 \times 0.13 = 22.6\ °c$ Ps= 2746 Pa

$\delta 2 = 24 - 11.1\ (0.13 + \frac{0.005}{1}) = 22.5°c$ 2730 Pa

$\delta 3 = 24 - 11.1\ (0.13 + \frac{0.005}{1} + \frac{0.02}{1.4}) = 22.3°c$ 2697 Pa

$\delta 4 = 24 - 11.1\ (0.13 + \frac{0.005}{1} + \frac{0.02}{1.4} + \frac{0.24}{0.99}) = 19.7°c$ 2299 Pa

$\delta 5 = 18 + 11.1\ (0.13 + \frac{0.005}{1}) = 19.5°c$ 2270 Pa

$\delta 6 = 18 + 11{,}1 \times 0.13 = 19.4\ °c$ 2256 Pa

The pressure inside the bathroom

δ air bath = 24°c , Ps = 2988 Pa , P= Ps. ϕ = 2390 Pa

δ air bath = 18°c , Ps = 2067 Pa , P= Ps. ϕ = 1654 Pa

the diffusion resistance (Air layer)

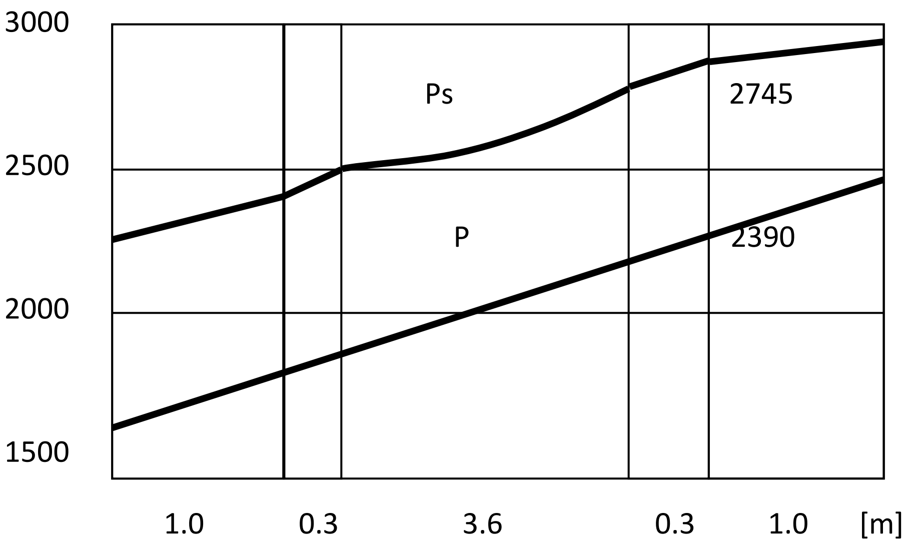
$\sum \mu.s = 0.005 \times 200 = 1.00$ m

$= 0.02 \times 15 = 0.3$ m

$= 0.24 \times 15 = 3.6$ m

$= 0.02 \times 15 = 0.3$m

$= 0.005 \times 200 = 1.0$ m



<u>There is no condensing because we don't have crossing between P and Ps</u>

3 – Condensing on the wall

We have condensing on the wall if Ps(δli) = Pli = ϕ . Ps(δli)

$\phi = \frac{Pli}{Ps(\delta li)} = \frac{2746}{2988} = 0.92 = 92\%$

the relative humidity is maximum 92%

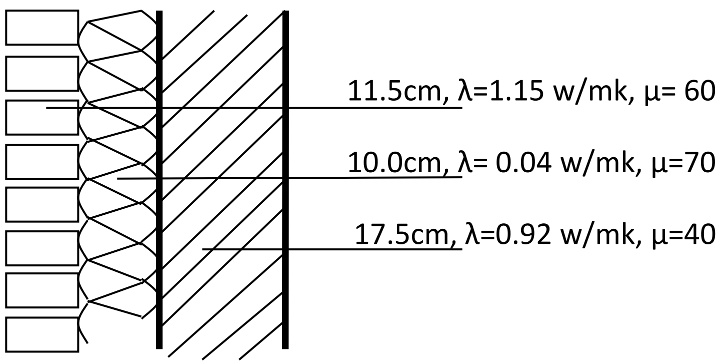
M = m .A = 0.119 x 7.5 = 0.89 g/h

A = 3 x 2.5 = 7.5 m²

$m = \frac{Pi1 - Pi2}{1.48 \cdot 10^{6} \cdot \sum \mu \cdot s} = \frac{2746 - 1654}{1.48 \cdot 10^{6} \cdot 6.2} = 0{,}119 \cdot 10^{-3}\, kg/m^2h = 0.199\, g/m^2h$

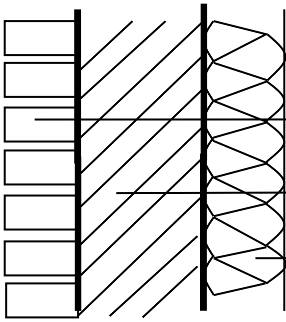
<u>Problem 23</u>

The outside wall of a villa is like the following sketch as showen.



Calculate

-Are we having condensing in the wall section?
-By the realization the building we have forget the isolation. And we build it like the following sketch. Are we having any changing in the temperature or in the moisture inside?



Solution

The testing of cond. Water:

q =k (δai – δao) = 0.34 (20+10) = 10.2w/m²

and

$$k = \left(\frac{1}{\alpha i} + \frac{S1}{\lambda 1} + \frac{S2}{\lambda 2} + \frac{S3}{\lambda 3} + \frac{1}{\alpha o} \right)^{-1}$$

$$= \left(0.13 + \frac{0.175}{0.93} + \frac{0.1}{0.04} + \frac{0.115}{1.15} + 0.04 \right)^{-1} = 0.34 \text{ w/m}^2\text{k}$$

<u>The Temperature in the wall</u>

$\delta li = \delta ai - q .(\frac{1}{\alpha i}) = 20 - 10.2 \times 0.13 = 18.7°c$ Ps = 2160 Pa

$\delta 1 = \delta ai - q .(\frac{1}{\alpha i} + \frac{s1}{\lambda 1}) = 16.7°c$ Ps = 1904 Pa

$\delta 2 = \delta ai - q .(\frac{1}{\alpha i} + \frac{s1}{\lambda 1} + \frac{s2}{\lambda 2}) = -8.8°c$ Ps = 289 Pa

$\delta lo = \delta ao + q \cdot (\frac{1}{\alpha o}) = -10 + 10.2 \times 0.04 = -9.6°c$ Ps = 269 Pa

<u>The humidity in and outside (winter)</u>

P inside = ϕ inside Ps = 0.5 x 2342 = 1171 Pa

P outside = ϕoutside Ps = 0.8 x 260 = 208 Pa

<u>The humidity in and outside (summer)</u>

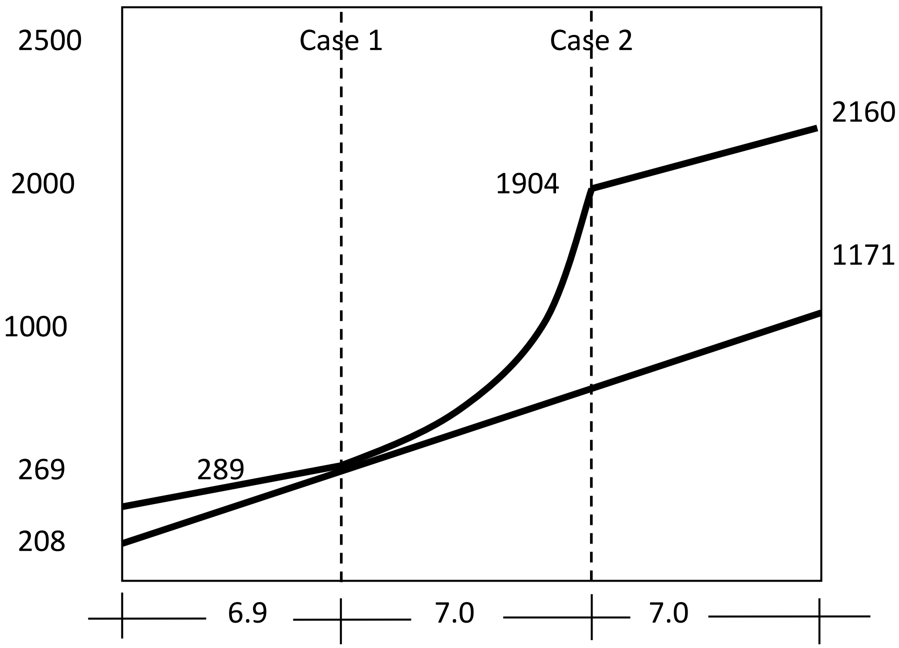
P= ϕ. Ps = 0.7 x 1404 = 983 Pa

<u>Diffusion resistance</u>

μ1 . s = 60 x 0.115 = 6.9m

μ2 . s = 70 x 0.1 = 7.0m

μ3 . s = 40 x 0.175 = 7.0 m



We have moisture in the layer between the Isolation and the covering outside.

2 – The temperature in the case 2

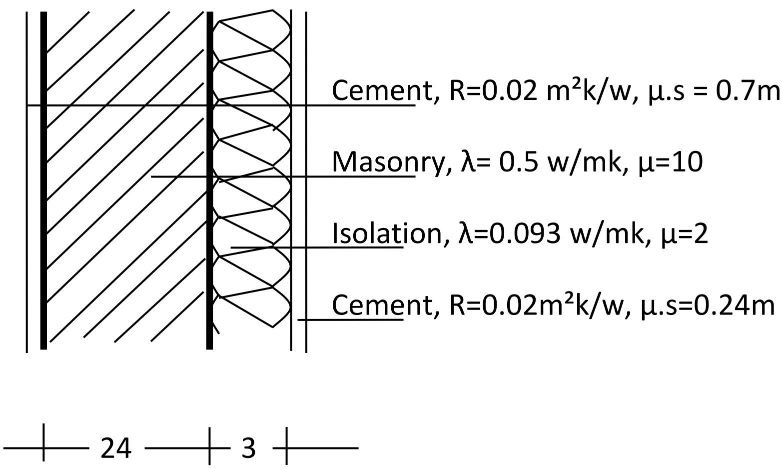
$\delta_{Isol.\ Layer/\ masonry} = \delta ai - q(\frac{1}{\alpha i} + \frac{s}{\lambda}) = 20\text{-}10.2(0.13+\frac{0.1}{0.4}) = -6.8°c$

Ps = 344Pa

We have moisture in the layer between the Isolation and the masonry wall.

Problem 24

The light weight structure of an outside wall is as the following sketch designed:



Air temperature	Inside 20°c
	Outside -15°c
Relative humidity	50%

Calculate

- Are we having condense water in the wall section?
- If yes in which layer?
- Calculate the water mass

Solution

The condensing on the surface of the wall exist if ps <p

$P = \phi \, . \, ps = 0.5 \times 2342 = 1171$ pa

$\delta li = \delta ai - k\,(\delta ai - \delta ao)\,\frac{1}{\alpha i} = 20 - 0.95\,(20+15)\,0.17 = 14.4°c$

$K = (0.17 + 0.02 + \frac{0.03}{0.093} + \frac{0.24}{0.5} + 0.02 + 0.04)^{-1} = 0.95 w/m^2k$

Ps (14.4°c) = 1643 pa > p , so there is no condensing on the surface of the wall.

The moisture in the wall section

Glaser Method

$Q = k\,(\delta ai - \delta ao) = 0.99\,(20+10) = 29.7\ w/m^2k$

$K = (0.13 + 0.02 + \frac{0.03}{0.093} + \frac{0.24}{0.5} + 0.02 + 0.04)^{-1} = 0.99 w/m^2k$

The temperature in the wall

$\delta li = 20-29.7(0.13) = 16.1°c$ Ps = 1833pa

$\delta 1 = 20 - 29.7\,(0.13+ 0.02) = 15.6°c$ Ps = 1775pa

$\delta 2 = 20 - 29.7\,(0.13+ 0.02 + \frac{0.03}{0.093}) = 6°c$ Ps = 936pa

$\delta 3 = 20 - 29.7\,(0.13+ 0.02 + \frac{0.03}{0.093} + \frac{0.24}{0.5}) = -8.3°c$ Ps = 302pa

$\delta lo = -10 + 29.7(0.04) = -8.8°c$ Ps = 289pa

Humidity

Winter P in = ϕ in . Ps = 0.5 x 2342 = 1171 pa

P out = ϕ out . Ps = 0.8 x 260 = 208 pa

Summer Ps = 1404 Pa

P = ϕ .Ps = 0.7 x 1404 = 983 pa

The water diffisoun according layers

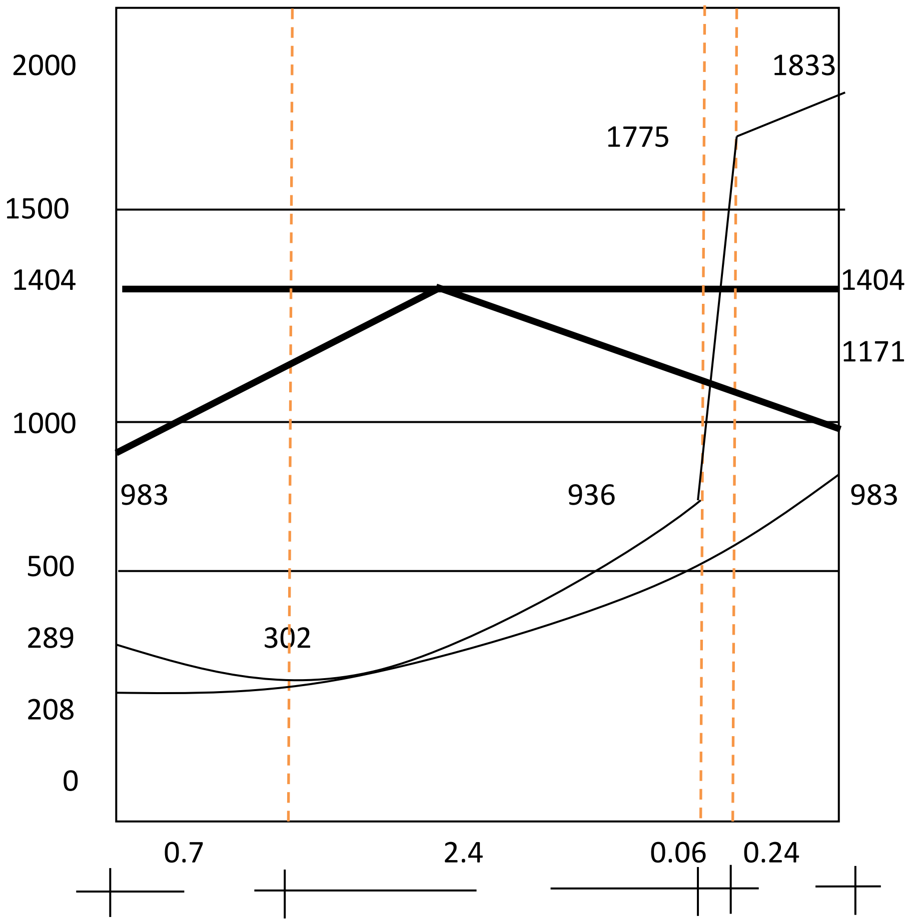
μ . s = 0.7m cement

= 2.4 m masonry

= 0.06 m Isolation

= 0.24 m cement

$\sum\mu.s = 3.4m$



The condense water mass

m c = t . m = 1440 .m [kg/m²]

$m = \frac{Pi - P2}{\frac{1}{\Delta i}} - \frac{P3 - Po}{\frac{1}{\Delta o}}$ [Kg/m²h]

$\frac{1}{\Delta i}$ = 1,48 . 10^6 (μ .s) = 1,48 . 10^6(0.24 + 0.06) = 1,48 . 10^6 . 0,3

$\frac{1}{\Delta o}$ = 1,48 . 10^6 (0.7) m²h Pa/Kg

$$m\,c = \frac{1440}{1.48 \cdot 10^6}\left(\frac{1171-936}{0.3} - \frac{302-208}{0.7}\right) = 0.632 \text{ kg/m}^2$$

The evaporation mass

$$m_e = \frac{Ps-Pi}{\frac{1}{\Delta i}} - \frac{Ps-Po}{\frac{1}{\Delta o}} \quad [\text{ Kg/m}^2\text{h }]$$

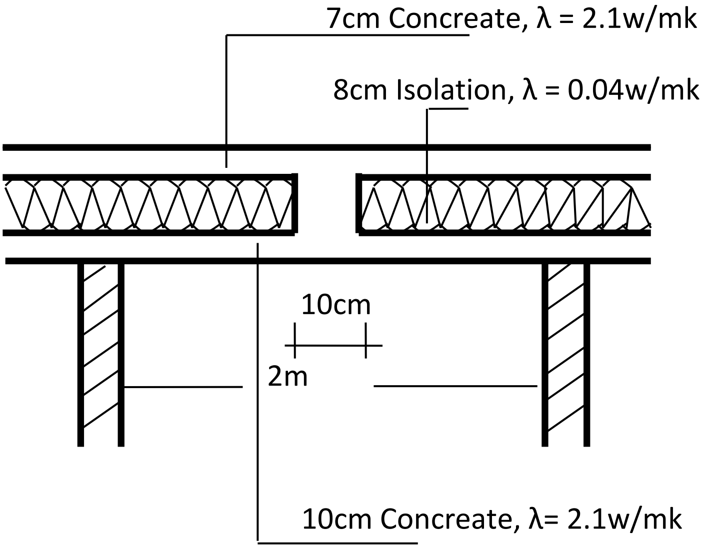
$$m_e = \frac{2160}{1.48 \cdot 10^6}\left(\frac{1404-936}{1{,}2+0.3} - \frac{1404-936}{1{,}2+0.7}\right) = 0.733 \text{ kg/m}^2$$

<u>**me > mc ⟶ no condensing**</u>

Which me: mass evaporation and mc: mass condensing

<u>Problem 25</u>

The Design of the outside wall is as a sandwich element as shown presented. Please check the building physics aspect in this design.



Room dimension	2.0 x 2.0 x 2.5 m
Gas constant for water diffusion	462 ws/kgk
Air temperature	inside 18°c
	Outside -10°c
Relative humidity	45%
Thermal resistance	Inside 0.17 m^2k/w
	Outside 0.04 m^2k/w

<u>Calculate</u>

- Sketch the temperature diagram in the two spaces?
- Are we have any condensing on the walls?
- Are we having any condensing in the wall section?

<u>Solution</u>

1- <u>Bridge</u>

$qb = Kb \cdot (\delta ai - \delta ao)$

$kb = (0.17 + \frac{0.25}{2.1} + 0.04)^{-1} = 3.04\ w/m^2k$

$qb = 3.04\ (28) = 85.1\ w/m^2$

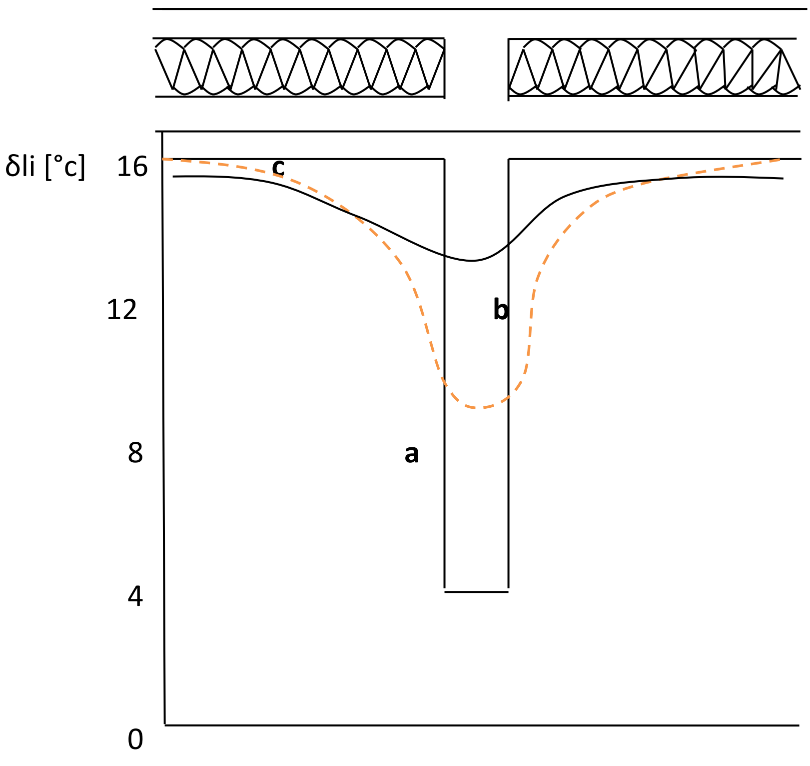
$\delta li = \delta ai - \frac{1}{\alpha i} q = 18 - 0.17 \times 85.1 = 3.5°c$

2- <u>Sandwich part</u>

$$ks = \left(0.17 + \frac{0.1}{2.1} + \frac{0.08}{0.04} + \frac{0.07}{2.1} + 0.04\right)^{-1} = 0.44\ w/m^2k$$

$$qs = Ks.\ (\delta ai - \delta ao) = 12.3\ w/m^2$$

$$\delta li = \delta ai - \frac{1}{\alpha i}\, q = 15.9°c$$



If ps > p then we don't have condense water on the wall .

P = ps (18°c) . 0,45 = 2067 x 0,45 = 930 Pa

Ps(bridge)= Ps(3.5°c) = 785 Pa

Ps (sandwich) = Ps(15.9°c) = 1809 Pa

In this case Ps < P then we have condensing on the wall.

The condensing in the wall section

$C = \frac{p}{R.T}$ [kg/m³]

$\Delta c = \frac{p-ps}{R.T} = \frac{930-785}{462.291} = 10.8 .10^{-3} kg/m^3 = 1.08\ g/m^3$

The condense water mass

$M = v .\Delta c$ = 2.5 x 2 x 2 x 1.08= 10.8 g

لا يجادل أحد في أننا نقدم بين صفحات هـذا الكتـاب كـل مـا هـو جديـد وقيم ويفيد في التقيم العلمـي لـلأداء الحـراري، بـل يتطـرق الأمـر إلـى التفريق بين الترسـيب فـوق سـطح الجـدار (رطوبـة هـواء Humidity) أو الترسـيب داخل الجدار (رطوبة مادة صلبة Mousiture).

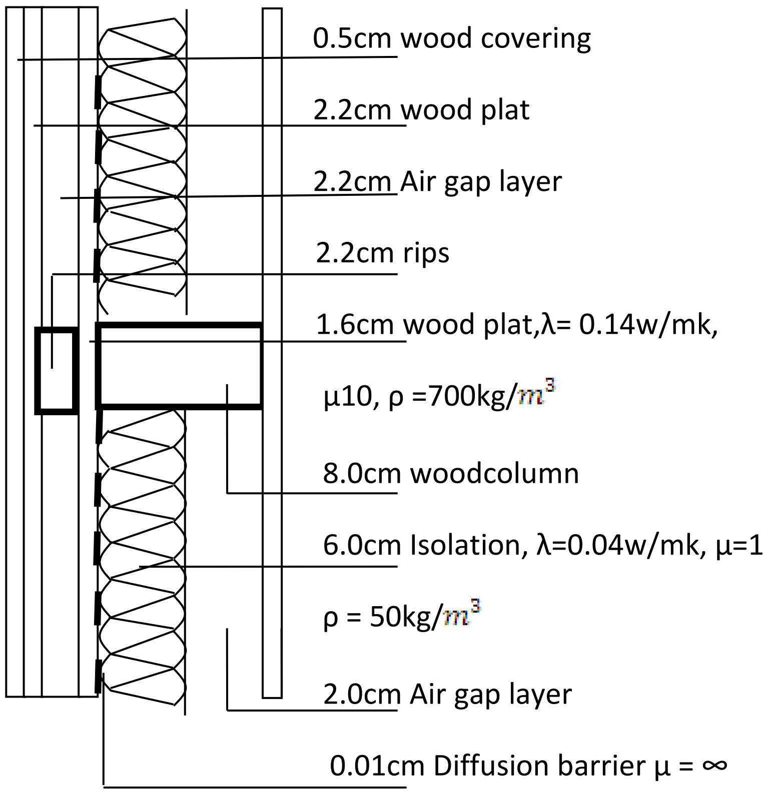
والمثـال الأخيـر – مثـال 25 – يفـرق بـين كميـة الميـاة المترسـبة فـي الحالتين. أذكر أنني وأنا أعمل في أحد المكاتب الألمانية العملاقة وكنت يومها أصمم مصانع لشركة Bayer كان لابد قبـل تسـليم المشـروع مـن تقديم Thermal Report لكل مشروع ، ولا تعطى الرخصـة لأي مشـروع صغر أم كبر إلا بالتقيم الحراري للمبنى.

الإختلاف بيننا وبينهم أنهم يستخدمون هذا التقيم الحراري في تصـميم التدفئة المركزية ونحن نستخدمه أو ينبغـي أن نسـتخدمه فـي التكيـف المركزي.

سؤال يطرح نفسه أو يحتمه السياق هـل إذا صـمم مهندسـي التكيـف الأعمـال المنوطـة بهـم دون مراعـاة لتصـميم الجـدارن بنـاء علـى الأداء الحراري لها يؤدي هذا إلى فقد في الطاقة الإجابة نعم.

Problem 26

The Design of the outside wall is as a sandwich element as shown presented. Please check the building physics aspect in this design.



Calculate

- Sketch the glaser diagram to check the condense water
- By which time can we have 100% humidity in the isolation layer

- Are we having any condensing in the wall section?
- Where can we put the diffusion isolation?

Solution

1- Glaser diagram

$q = k.\,(\delta ai - \delta ao) = 0.48\,(20+10) = 14.4 w/m^2$

$k = (\,0.13 + \frac{0.016}{0.14} + 0.14 + \frac{0.06}{0.04} + \frac{0.016}{0.14} + 0.08)^{-1} = 0.48\ w/m^2k$

The Temperature

$\delta li = \delta ai - q\,.\,\frac{1}{\alpha i} = 20 - 14.4 \times 0.13 = 18.1°c$ Ps=2080 pa

$\delta 1 = 20 - 14.4\,(0.13 + \frac{0.016}{0.14}) = 16.5°c$ Ps=1880 pa

$\delta 2 = 20 - 14.4\,(0.13 + \frac{0.016}{0.14} + 0.14) = 14.5°c$ Ps=1654 pa

$\delta 3 = -10 + 14.4\,(0.08 + \frac{0.016}{0.14}) = -7.2°c$ Ps=332 pa

$\delta lo = -10 + 14.4\,(0.08) = -8.9°c$ Ps=286 pa

Layers	Temperature	Sat. pressure	pressure
Air inside	20°c	2342 pa	1171 pa
Level inside	18.1°c	2080 pa	1171 pa
Wood plat	16.5°c	1880 pa	
Isolation	14.5°c	1654 pa	
Wood plat	-7.2°c	332 pa	
Level outside	-8.9°c	286 pa	208 pa

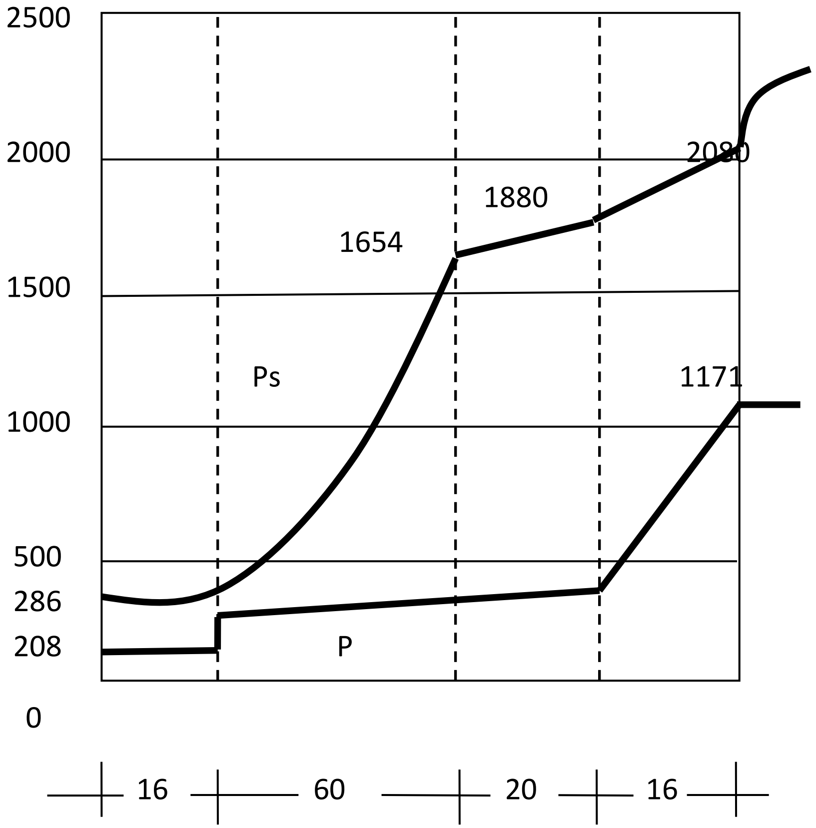
Air outside	-10.0°c	260 pa	208 pa

<u>The condense water mass</u>

M = $\frac{p-ps}{1.48.10^6 \ \sum\mu.s}$ = $\frac{1171-332}{1.48.10^6.1.68}$ = 3.374 . 10^{-4}kg/m²h

P = ϕ. Ps(20°c) = 1171 pa

Ps condensing barrier = Ps(-7.2°c) = 332 pa



$\sum (\mu.s)i = 100.\ 0{,}016 + 0{,}02\ .1 + 0{,}06\ .\ 1 = 1.68$

$m = \frac{p - p\ evap}{\frac{1}{\Delta 1}}$ [kg/m²h]

$\frac{1}{\Delta 1} = 1{,}48\ .\ 10^{6}\ .\ \mu\ .s = 1{,}48\ .\ 10^{6}.\ 1.6$ m²h/kg

$P\ evap = p - m\ .\ \frac{1}{\Delta 1} = 1171 - 3.374\ .\ 10^{-4} \times 1.48\ .\ 10^{6} \times 1.6 = 372$ pa

$M = \frac{p - p\ evap2}{\frac{1}{\Delta 1} + \frac{1}{\Delta 2}}$ [kg/m²h]

$\frac{1}{\Delta 2} = 1.48\ .\ 10^{6} \times 0.02$ m²h/kg

$P\ evap2 = p - m\ (\frac{1}{\Delta 1} + \frac{1}{\Delta 2}) = 1171 - 3{,}374\ .10^{-4} \times 1{,}48\ .\ 10^{6}(1.6 + 0.02)$

$= 362$ pa

<u>The timing to have 100% humidity in the isolation layer</u>

$U = \frac{m\ H2o}{m} = \frac{m\ .t}{d\ .\varphi} = \frac{t\ .\ \ 3{,}362\ .10^{-4}}{0{,}06\ .\ \ 50} = 1{,}12\ .10^{-4}t$

If u 100% =1

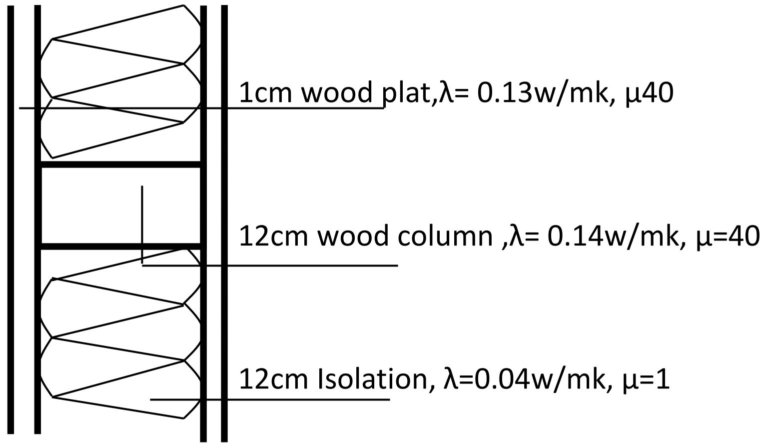
$T = \frac{1}{1{,}12\ .10^{-4}} = 8929$ h = 372 day = <u>1 year</u>

$U_{m}60 = 1{,}12\ .\ 10^{-4}\ \ .\ 1440 = 0.161$

<u>We have to put the diffusion barrier sheet on the worm layer</u>

Problem 27

In a high wormed room the design of the partition looks like the shown drawing. Please check the building physical phenomena.



Room dimension	2 x 2 x 2,5m
Gas constant	462 ws/kgk
Air temperature	24°c
Relative humidity	60%
Thermal resistance	0.17 m^2k/w

Calculate

- The diffusion of the water in the space as a mass
- If we have air temperature of 80°c and in the entrance 24°c can condensing water existing in the wall section
- By which relative humidity are we have condensing and where

- If we have $3g/m^2$ as a water diffusion mass on the wall surface are we have condensing in the wall section?

<u>Solution</u>

1- Water concentration in the air

$C = \frac{m\ H2o}{v} = \frac{P}{R\ .T}$ [kg/m³]

P = ϕ . ps = ϕ . ps(24°) = 0,6 . 2988 pa = 1793 pa

T = 24 + 273 = 297 k

$C = \frac{1793}{462\ .297}$ = 0.013 kg/m³ = 13 g/m³

2- The condense water- moisture

q = K (δai – δao) = 0.29 (80-24) = 16.2 w/m²

$k = (\frac{1}{\alpha i} + \sum\frac{s}{\lambda} + \frac{1}{\alpha o})^{-1} = (0.17 + 2.\frac{0.01}{0.13} + \frac{0.12}{0.04} + 0.17)^{-1} =$ 0.29w/m²k

The Temperature in the wall section

$\delta li = \delta ai - q\,\frac{1}{\alpha i} = 80 - 16.2\ (0.17) = 77.2°c$

$\delta 1 = \delta ai - q\ (\frac{1}{\alpha i} + \frac{s}{\lambda}) = 80 - 16.2\ (\ 0.17 + \frac{0.01}{0.13}\ ) = 76°c$

$\delta 2 = \delta ao + q\ (\frac{1}{\alpha o} + \frac{s}{\lambda}) = 24 + 16.2\ (\ 0.17 + \frac{0.01}{0.13}\ ) = 28°c$

$\delta lo = \delta ai + q\,\frac{1}{\alpha o} = 24 + 16.2\ (0.17) = 26.8°c$

The saturated pressure

$$Ps = C1 \exp\left[\frac{C2\,.\delta}{C3+\delta}\right]$$

Constant	Temperature / water	Temperature / frost
C1	610.7	610.7
C2	17.08	22.44
C3	234.1	272.4

$Ps(\delta) = 610.7 \exp \frac{17{,}08\,.\delta}{234{,}18+\delta}$ [pa]

$Ps(80) = 610{,}7 \exp\frac{17{,}08\,.80}{234{,}18+80} = 47280$ pa

$Ps(77.2) = 610{,}7 \exp\frac{17{,}08\,.77.2}{234{,}18+77.2} = 42165$ pa

$Ps(76) = 610{,}7 \exp\frac{17{,}08\,.76}{234{,}18+76} = 40121$ pa

$Ps(26.8) = 610{,}7 \exp\frac{17{,}08\,.26.8}{234{,}18+26.8} = 3785$ pa

Ps(24) = 2988pa ; p = 1793 pa

<u>**In our case**</u>

$$\frac{pli-plo}{\sum\mu.s} = \frac{p3-pai}{0.4}$$

$Pli = \frac{0.92}{0.4}$ (3785 – 1793) + 1793 = 6375 pa

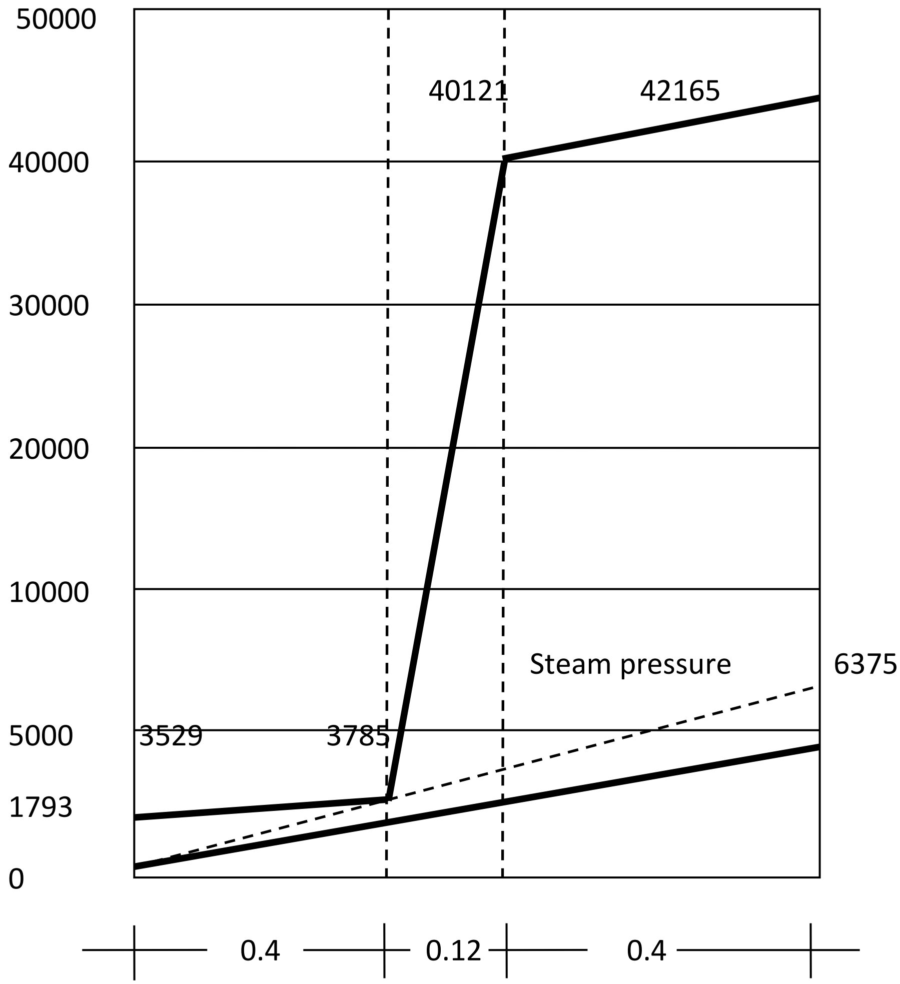
$\varphi = \frac{p}{ps} = \frac{6375}{42165} = 0.15 = 15\%$

By 3g/m² $\quad m = \frac{pi - po}{\frac{1}{\Delta}} = 3$ g/m²

$P = 0.003 \frac{1}{\Delta} + Po = 0{,}003 \times 1{,}36 .10^{6} + 1793 = 5873$ pa

$\phi = \frac{5873}{47280} = 0.12 = 12\%$

No condensing because $\phi < 15\%$



في المثالين السابقين

-يمكننا حساب الضغط الذي سيتم عنده تبخر المياة المترسبة

-الزمن الذي سيصل معه نسبة الرطوبة إلى 100%

-نسبة تركيز بخار الماء في الهواء

-حساب الضغط المشبع بمعطيات جديدة

Problem 28

Two persons produce 100g water diffusion in the space. We have air changing through the opening of 0.5 and the air temperature of20°c.

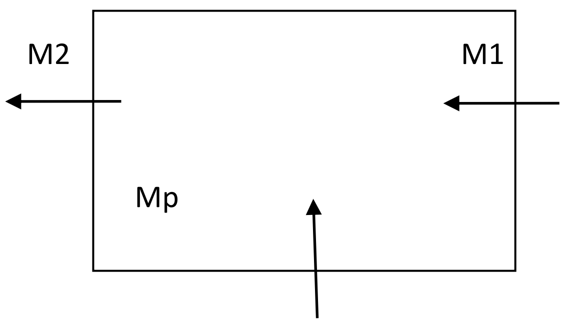
The room dimension is 5 x 4 x 2.5 and the gas constant 462ws/kgk

Calculate

- The relative humidity if the relative humidity outside 80% and the temperature between -15°c till +20°c.
- Which temperature must we have on the wall to avoid the condense water
- By which outside temperature are we have condensing if the conductivity 1,39w/m²k

Solution

To calculate the relative humidity we must build the humidity balance



M1 + Mp –M2 =0

Which

M1 = co . v . nl [kg/h]

M2 = ci . v . nl [kg/h]

Mp =0.1 kg/h

V . nl (Co –Ci) + Mp = 0

Ci = Co + $\frac{Mp}{v.n}$ [kg/m³]

Ci = $\frac{\varphi i .Ps\ (\delta ai)}{R.Tai}$ [kg/m³]

Co = $\frac{\varphi o .Ps\ (\delta ao)}{R.Tao}$ [kg/m³]

$\frac{Mp}{V.nl} = \frac{0.1}{(4\ x\ 5\ x\ 2.5).0.5}$ = 4.10^{-3} kg/m³

$\frac{\varphi i .Ps\ (\delta ai)}{R.Tai} = \frac{\varphi o .Ps\ (\delta ao)}{R.Tao}$ + 4.10^{-3}

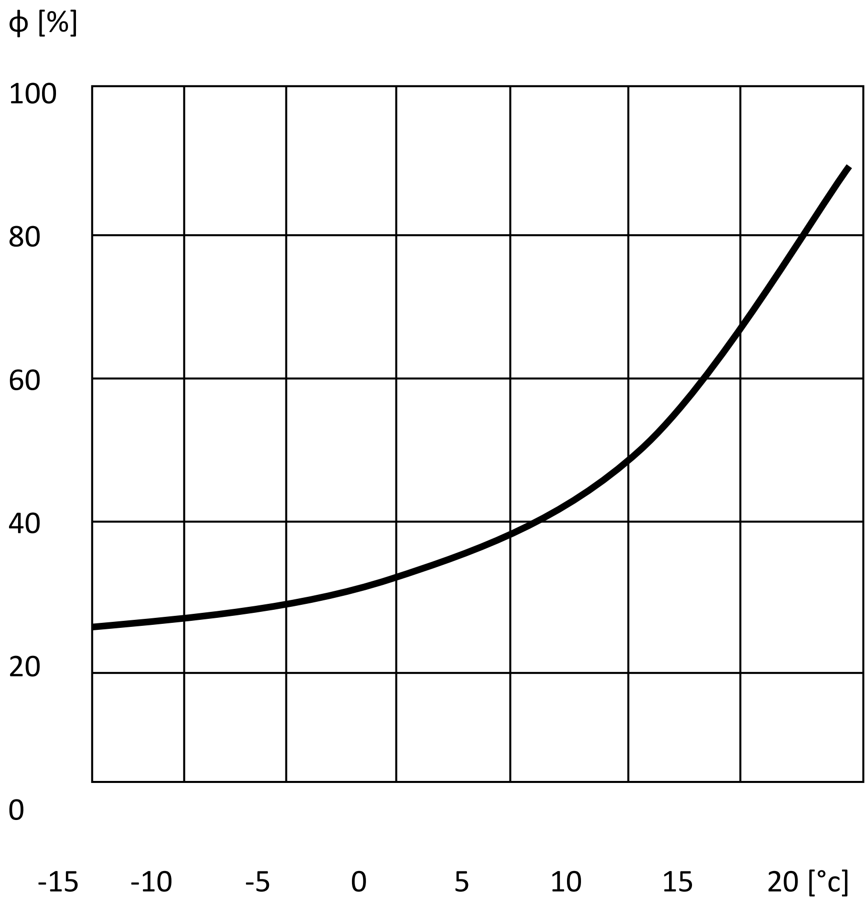
ϕ = $\frac{Tai}{Ps\ (\delta ai)}$ ($\frac{\varphi o .Ps\ (\delta ao)}{R.Tao}$ +4.10^{-3}.R) [%]

Tai =20 + 273 = 293 k

ϕ = $\frac{293}{2342}$ ($\frac{0.8 .Ps\ (\delta ao)}{R.Tao}$ +4.10^{-3}.462) = 0.125 ($\frac{0.8 .Ps\ (\delta ao)}{R.Tao}$ + 1.85).100 [%]

δ ao	-15	-10	-5	0	5	10	15	20
Tao	258	263	268	273	278	283	288	293
Ps(δai)	165	260	401	611	873	1229	1708	2342
ϕ	30	33	38	45	55	67	82	100
Ps(δli)	703	773	890	1054	1288	1569	1920	2342

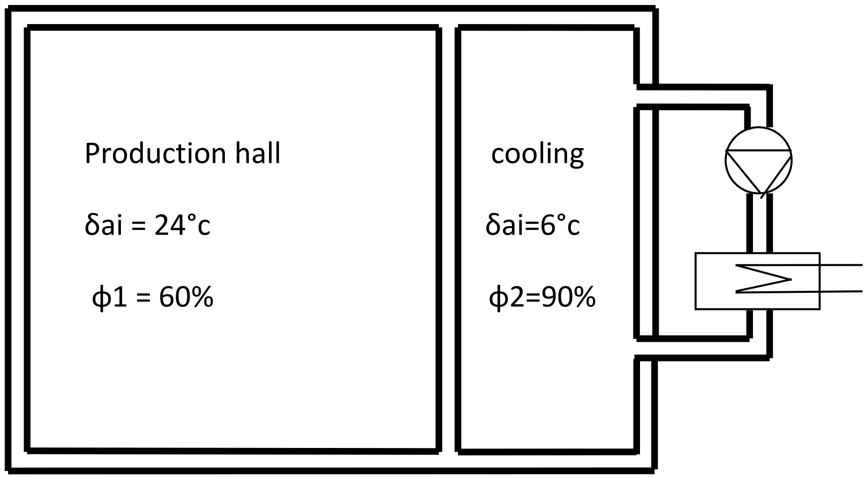
The graphic of the humidity is in relation with the temperature outside



The temperature on the wall must equals with the temperature inside

Problem 29

In this example we have two rooms with different function. We have also air changing power 10. The cooling machine reduce the temperature from 6°c to 2°c.



Cooling room 4.00 x 4.00 x 2.50

Separation wall 24cm $\lambda = 0.99$

Heat capacity in the air 0.35 wh/m³k

Gas constant 462 ws/kgk

Calculate

- The cooling energy if the transmission of heat is 500w

- The separation wall between the two rooms must be isolated. which thickness of isolation are we have if the conductivity 0.04w/mk
- Which water diffusion are we have if the separation wall only 5% from the total amounf.

Solution

The cooling energy

Q total = Qu + Km .Am .Δδ = 500w $\left(\frac{2}{\alpha i} + \frac{S}{\lambda}\right)^{-1}$x 4 x 2.5 (24-6)

= 500w (2 x 0.13 + $\frac{0.24}{0.99}\Big)^{-1}$x 4 x 2.5 (18) =858.3w

Q cooling = nl . ρl . c .v . Δδ = 10 x 0,35 (4x4x2.5)(6-2) = 560w

Qwall = Q cooling - Qu = kw.AM .Δδ = 560-500 = 60w

Kw = $\frac{Qw}{AM\ .\Delta\delta}$ = $\left(\frac{1}{\alpha i} + \frac{1}{\Lambda m} + \frac{1}{\Lambda D} + \frac{1}{\alpha i}\right)^{-1}$ = $\frac{60}{10.(\ 24-6)}$ = 0.33 w/m²k

$\frac{1}{\Lambda D}$ = $\frac{1}{kw}$ - 2$\frac{1}{\alpha i}$ - $\frac{1}{\Lambda m}$ = $\frac{1}{0.33}$ – 2 x 0.13 – $\frac{0.24}{0.99}$ = 2.53m²k/w

$\frac{1}{\Lambda D} = \frac{s}{\lambda}$

S = $\frac{1}{\Lambda D}$. λ = 2.53 x 0.04 =0.1m

The water diffusion

Cv = $\frac{\varphi .Ps\ (6°c)}{R.T}$ = $\frac{0.9\ x\ 936}{462\ x\ 279}$ = 6,54 . 10^{-3}kg/m³ = 6,54g/m³

$Ca = \frac{\varphi . Ps\ (2°c)}{R.T} = \frac{706}{462\ x\ 279} = 5,56\ .\ 10^{-3}$kg/m³ = 5,56g/m³

M = V . n . Δc = 40 .10 (6.54 -5.56) = 392 g/h

<u>Water diffusion through the separation wall</u>

M1 = 0.005 x 392 = 1.96 g/h

<u>Water diffusion intensity</u>

$M2 = \frac{m1}{A} = \frac{1.96}{10} = 0.196 g/m^2h$

$m' = \frac{p1-p2}{\frac{1}{\Delta w}} = \frac{0.6\ x\ 2988 - 0.9\ x\ 936}{\frac{1}{\Delta}} = 0.196\ g/hm^2 = 1.96\ .10^{-4} kg/m^2h$

$\frac{1}{\Delta w} = \frac{0.6\ x\ 2988 - 0.9\ x\ 936}{1.96\ .10^{-4}} = 4.85\ .\ 10^{6} m^2hPa/kg$

For the layers in the separation wall

$\sum(\mu.s) = \frac{4.85\ .10^{6}}{1.48\ .10^{6}} = 3.28m$

$\sum(\mu.s) = (\mu.s)M + (\mu.s)D = 3.28m$

$(\mu.s)M = 3.28 - (\mu.s)D = 3.28 – 1 \times 0.1 = 3.18m$

$\mu M = \frac{3.18}{S} = \frac{3.18}{0.24} = 13.3$

<u>Light concrete has μ=13 and λ=0.99</u>

<u>To check the condensing we must use glaser method</u>

<u>Alternative 1: Isolation in the production hall</u>

$q = k(\delta 1-\delta 2) = k(\delta ai – \delta ao) = 0.33 (24 - 6) = 5.9 w/m^2$

$\delta ai = 6°c$

$\delta li = \delta ai + q\frac{1}{\alpha i} = 6 + 5.9(0.13) = 6.8°c$

$\delta 1 = \delta ai + q(\frac{1}{\alpha i} + \frac{1}{\Delta M}) = 6 + 5.9(0.13+\frac{0.24}{0.99}) = 8.2°c$

$\delta lo = \delta ai + q(\frac{1}{\alpha i} + \frac{1}{\Delta M} + \frac{1}{\Delta D}) = 6 + 5.9 (0.13+\frac{0.24}{0.99} + 2.53) = 23.1°c$

$\delta ao = 24°c$

<u>Alternative 2 : Isolation in the cooling room</u>

$\delta ai = 6°c$

$\delta li = 6.8°c$

$\delta 1 = \delta ai + q(\frac{1}{\alpha i} + \frac{1}{\Delta D}) = 6 + 5.9 (0.13+ 2.53) = 21.7°c$

$\delta lo = 23.1°c$

$\delta ao = 24°c$

<u>The saturated pressure</u>

$\delta ai = 6°c$ $Psai = 936Pa$ $P=0.9 \times Ps = 842Pa$

$\delta li = 6.8°c$ $Psli = 989Pa$

$\delta 1 = 8.2°c$ $Ps1 = 1089Pa$

$\delta lo = 23.1°c$ $Pslo = 2831Pa$

$\delta ao = 24°c$ $Psao = 2988Pa$ $P=0.6 \times Ps =1793Pa$

<u>The saturated pressure (alternative 2)</u>

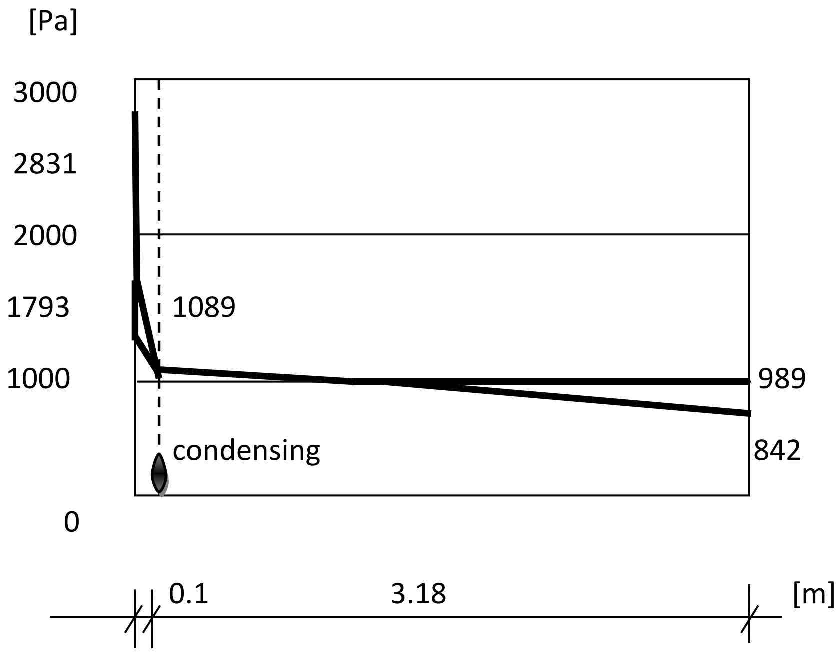
$\delta ai = 6°c$ $Psai = 936Pa$ $P=0.9 \times Ps = 842Pa$

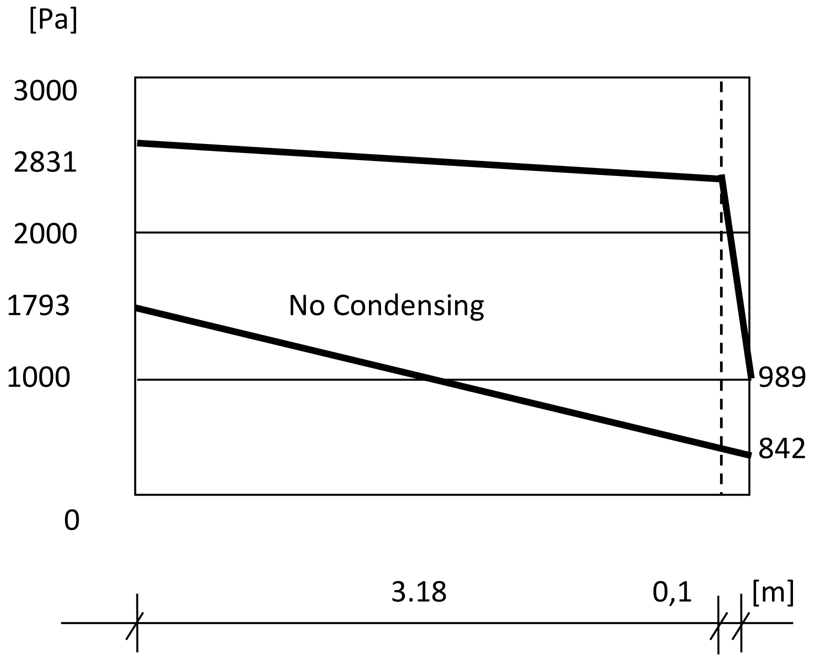
$\delta li = 6.8°c$ $Psli = 989Pa$

$\delta 1 = 21.7°c$ $Ps1 = 2600Pa$

$\delta lo = 23.1°c$ $Pslo = 2831Pa$

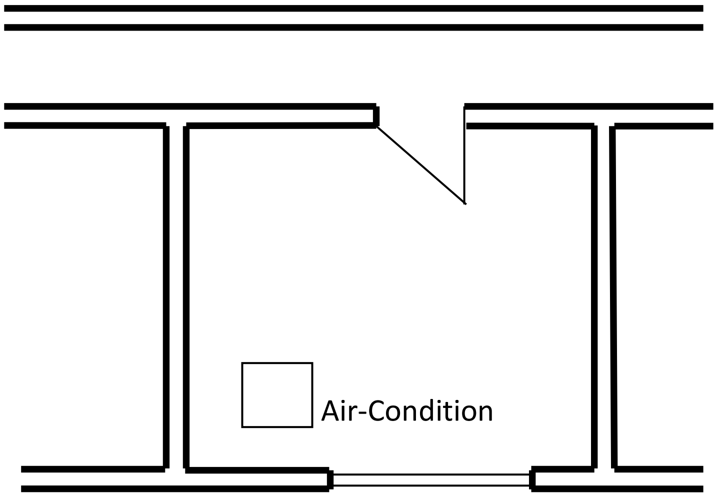
$\delta ao = 24°c$ $Psao = 2988Pa$ $P=0.6 \times Ps =1793Pa$


[Pa]
3000
2831
2000
1793
1089
1000
989
condensing
842
0
0.1
3.18
[m]


[Pa]
3000
2831
2000
1793
No Condensing
1000
989
842
0
3.18
0,1
[m]

Problem 30

One outsider room must have through air-conditioned the temperature of 18°c. The temperature of the corridors and the others rooms is also 18°c. And the relative humidity is 60%



Room dimension 6 X 4 X 2.5m

Windows area 2m² μ= ∞

Outside wall	1.5cm cement, μ = 10
	36,5cm masonry, μ = 10
	2.5 cm cement, μ = 20

Inside wall	1 cm cement, μ = 8
	5 cm isolation, μ = 1

	1 cm cement, μ = 8

Gas –constant 462ws/kgk

Air-changing units $0.1h^{-1}$

Temperature	inside 18°c
	Outside 25°c

Relative humidity	inside 60%
	Outside 50%

Calculate

- The relative humidity in the space inside ?

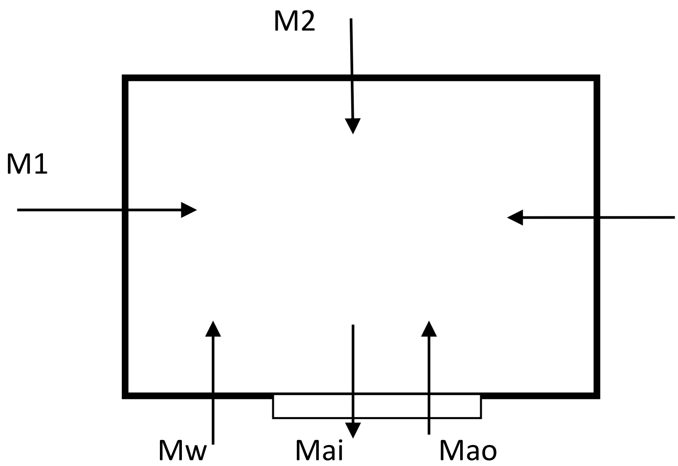
Solution

Relative humidity

$\Phi = \frac{p}{ps}$

With P = Ps(18°c) = 2067 Pa

The humidity balance



$\sum M = Mw + Ma + M1 + M2 + M3 = 0$

With $M = m.A$ [kg/h]

The humidity intensity

$m = \frac{\Delta p}{\frac{1}{\Delta}}$ [kg/m²h]

The humidity resistance

$\frac{1}{\Delta} = 1{,}48 \cdot 10^{6} \sum (\mu \cdot s)$ m²hpa / kg

Ma = Mai –Mao

With

Ma = (Ca-Ci) .V.nl [kg/h]

And

$Ca = \frac{Pao}{R - Tao} = \frac{1586}{462 \times 298} = 11{,}52 \cdot 10^{-3}$ kg/m³

$C_i = \frac{P_{ai}}{R - T_{ai}} = \frac{P_{ai}}{462 \; x \; 291} = 7{,}4 \; . \; 10^{-6} P_{ai}$ [kg/m³]

$M_a = (11.52 - 0.0074 P_{ai}) \; . \; 10^{-3}$ (6 x 4 x 2.5) 0.1 [kg/h]

Outside wall $\sum(\mu \; . \; s) = 0.365 \times 10 + 0.025 \times 20 + 0.015 \times 10 = 4.3$m

$\frac{1}{\Delta} = 1.48 \; . \; 10^{6} \times 4.3 = 6.36 \; .10^{6}$ m²h pa /kg

inside wall $\sum(\mu \; . \; s) = 0.01 \times 8 \times 2 + 0.05 \times 1 = 0.21$

$\frac{1}{\Delta} = 1.48 \; . \; 10^{6} \times 0.21 = 0.31 \; .10^{6}$ m²h pa /kg

$P_{so} = P_s(25°c) = 3172$ Pa , $P_{ao} = 0.5 \times 3172 = 1586$ Pa

$P_{s1} = P_{s2} = P_{s3} = P_s(18°c) = 2067$ Pa ,

$P_1 = P_2 = P_3 = 0.6 \times 2067 = 1240$ Pa

$M_w = \frac{P_{ao} - P_{ai}}{\frac{1}{\Delta}} \; . \; A_w = \frac{1586 - P_{ai}}{6.36 \; .10^{6}} (6 \times 2.5 - 2) = 3{,}24.10^{-3} - 2{,}0 \; .10^{-6}$ Pai

$A_{wall} = (6 \; . \; 2{,}5) - A_{window} = 13m^2$

$M_1 + M_2 + M_3 = \frac{P_{a1} - P_{ai}}{0.3 \; x \; 10^{6}} \; . \; (6 \times 2.5 + 2 \times 4 \times 2.5)$ [kg/h]

$= \frac{1240 - P_{ai}}{0.31 \; x \; 10^{6}} \; . \; 35 = 0.14 - 0.113 \; . \; 10^{-3}$ Pai

$M = [(3.24 - 0.002. \; P_{ai}) + (140 - 0.113 P_{ai}) + (69.1 - 0.044 P_{ai})] \; .10^{-3} = 0$

$P_{ai} = \frac{3.24 + 140 + 69.1}{0.002 + 0.113 + 0.044} = 1336$ Pa

$\Phi = \frac{P_{ai}}{P_s \; (18)} = \frac{1336}{2067} = 0.646 = 64.6\%$

Buildings behavior under thermal

The advances of high tech in the building sector, intelligent buildings, air-conditioning systems, new materials and buildings technologies, including glass technologies, have opened great possibilities in realizing the importance of and expressing architectural vision, imagination, new ideas, but have also limiting energy aspects, ecological situation, environment protection and sustainability or green building directions. The new architectural era, with computer modeling and building simulation have enabled us to analyze a building in its real life, predicting its dynamic behavior and estimating its energy consumption, indoor air quality, lighting, even in the projecting period, when buildings design is in its initial phase.

Form a "static mass" the architecture produces "adaptive" building structure, with an envelope as a skin moderating heat flows. A building presents a fully integrated, intelligent adaptable structure, both in terms of the used materials, their fabric, locations, information technologies and all building operation systems. With the central control system, buildings intelligence and with defaulted values concerning energy systems, buildings are getting characteristic of a human body, at least in regard to the reaction to thermal conditions – but in which degree?

In a cold environment, a human body by lowering blood circulation toward the skin surface by the blood vessels tightening, thus conserving

Body's heat and controlling its heat losses. That is an instinctive reaction, default attribute, without any conscious possibility to

influence it. But a man can improve the condition by dressings, adding thus an additional protective cover over his body, above his skin, and acquiring thus better insulation. This human way of additional protection can be applied on buildings.

Why can't they be covered with movable covers, in winter to protect them from the wind and low outside temperature, and in summer from the sun, in order to reduce heat gains from the solar radiation, and so reduce the necessary energy for its air conditioning? Maybe a kind of automatic lowered shades? Or covering a building with a second façade?

Double Facades

There are buildings today with such doubled protection, mostly static like a "pullover" or a" winter coat" on a man. Those are the double façade buildings- additional cover is added to the main façade, usually made of glass. A double façade in the summer could be protection from the sun.

A winter coat or a pullover is changed by a light material blouse of the bigger size, roomy over the body, so that the air may circulate next to the body and cool it, enabling the body to transfer a heat out of a body.

In the double façade , various forms of shades, curtains or similar devices, are put in the inter-space for the sun-protection , though a passage must be provided for the outside air circulation, so that the inter- space temperatures may be as close as possible, if not identical, with the outside temperature.

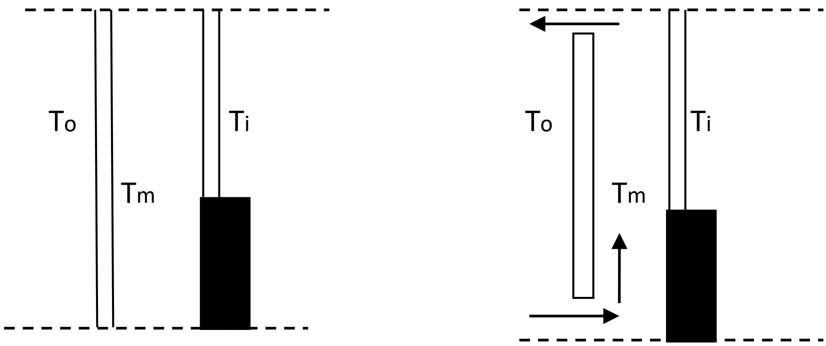


Fig:1.12 Double fasades alternatives

From the constructional point of view, a double-façade outer envelope may be continuously extended by covering the total height of a building, or discontinued with breaks at each floor level. Disregarding the height, the inter-space in opened both at the bottom and at the top, thus providing the outdoor air circulation in the summer, when the temperature of inter-space should be as low as possible, in principle equal to the outside temperature. Or the openings my be closed, which is case during the winter, in order to trap the air in the inter-space, which will act as insulation layer , with the temperature above the outside temperature, producing lower heat losses of a building

The diagram shows the course of temperatures in the inter-space on an average sunny day in January, for the south-turned double façade, in Belgrade (45NL). The figure right presents the temperatures during

the sunny day in July. The outside temperature in given as well on both figures.

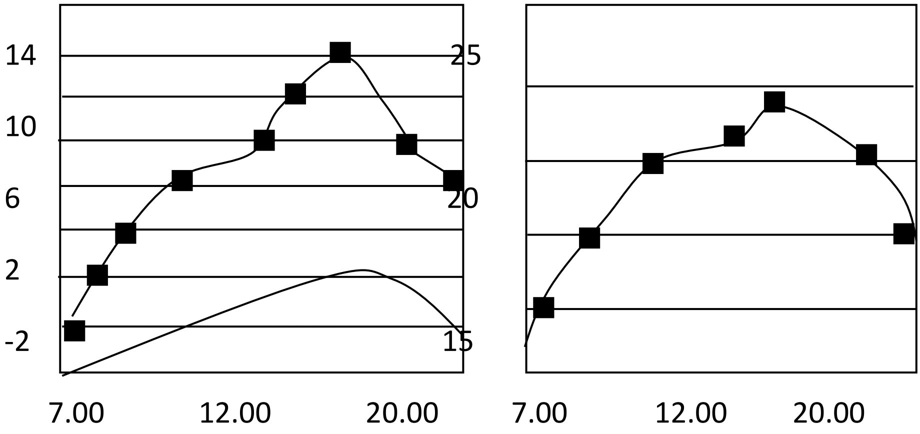


Fig:1.13 The diagram shows the course of temperatures in the inter-space on an average sunny day in January

It is evident that during the heating period, cavity temperature of a building with a double-facade is above the outside one, and will have lower heat losses and decreased needs for heating. In the summer, during the cooling period, the temperature between the two facades could be equal or very close to the outside temperature, and additionally can have smaller heat gains from solar radiation, depending on glass properties regarding solar transmittance as a consequence, heat gains will not be above the gains of single façade buildings.

During the summer, one uses his conscious reactions for the additional protection of his body. He may protect himself by hats, or make a shade using a parasol. Similar protection is used in buildings by various curtains, shades, and venetian blinds on windows, while today copies of caps and parasols are constructed, as immovable

elements over roofs, or movable, dependingon the sun temporary location. All those protections may be also used on facades.

Examples of building protection from the solar radiation are numerous, especially in the regions of tropical conditions. An illustrative example is a building designed by the English architect Grimshow in Seville, built for the EXPO 1992.

Movable protection on the roof is put according to the momentary sun location, controlled by buildings intelligence

<u>Buildings evaporative cooling</u>

During the summer, heat enters into buildings from the outside through hot air and solar radiation, but there are also heat gains inside (lighting, domestic hot water systems, people, electric appliances and devices). Such heat must be eliminated so that the inside temperature would not be

Above the planned one, for example 22°c. In conditions when the outside temperature is above the human body temperature, the only way for a man to eliminate his inner heat is by perspiration, through evaporation. A building cannot sweat, so that it has to be cooled mechanically by air conditioning system.

But can we use the human body sweat evaporation effect on buildings? There are buildings for which it may be said that they use the effect of water evaporation for their cooling, as in the case of a mans sweating.

It is an old practice to put water sprinklers on the roofs of large surface, and use them at high out side temperatures, when the sun radiates intensity. The roof is so moistened, and because of the heat absorbed by the outer roof surface and the air layer next to it, water evaporation occurs, and the roof temperature is lowered. Pools are also installed on roofs of multi-story residential and business buildings.

The idea to let the water flow down a building façade, an imitation of human sweating, is an option in a modern architecture, in case of glass facades as frequently used elements in natural reaction. Flowers, grass, but also water as an especially important element in some cultures becomes a repeatedly used elements in the modern architectural expression-most often inside the large halls, restaurants, atriums. There are several buildings around the world with water flowing down the glass vertical or the inclined façade. One of such examples is again the building of the British pavilion in Seville. Such façade has smaller coefficient of the solar radiation transmission , and with the water layer, due to its evaporation, the temperature next to the façade is significantly lower than the outside one, reducing so the heat gain from the solar radiation, as well as from temperature difference between the outside and the inside.

The measurements provided above the glass with an angle of 45 degrees are presented on the fig. showing the outside temperature, the temperature of dry and wet glass, solar radiation intensity through dry and wet glass. Uniform water flow above glass façade has a lower solar radiation transmittance for 10-15%than an ordinary dry glass. And when the water flow is turbulent and disturbed, even 25-30% depending on water quantity.

The temperature of a glass under water flow was about 10°c lower than the glass without water above it. The temperature on a water

inlet and outlet was very small, as the distance between them was 1.5m only.

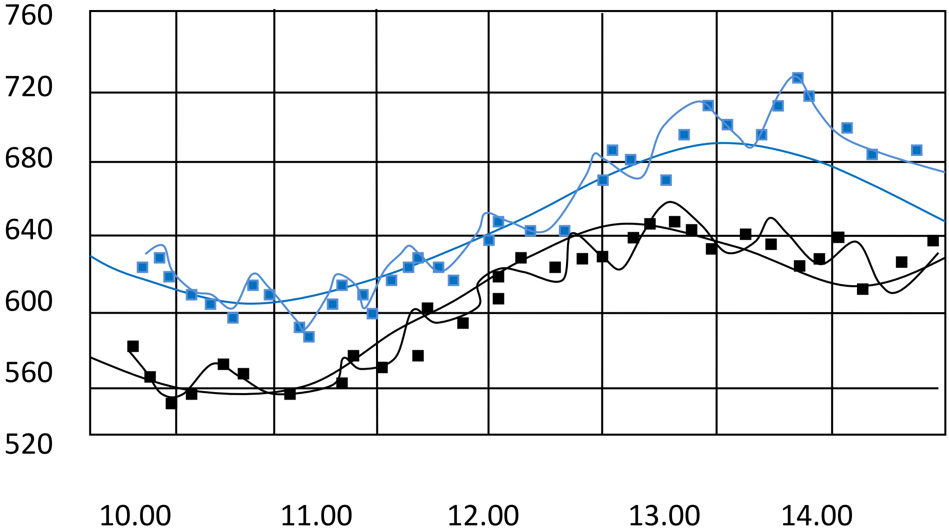


Fig:1.14 The figure showing the outside temperature, the temperature of dry and wet glass

Developing and upgrading the cooling process of the British Pavilion

The weekniss of the solution of Grimshow was that the cooling processes were in the wall and not in the double layering roof. If we compare between grimshow solution and the solution from the author, we find that the temperature between inside and outside rise up from 8 to 17,3 °c.

The second reason for this pheonmen is the evaporation. The system from grimshow is closed but the system from the author allows the evaporation by contacting between air and water.

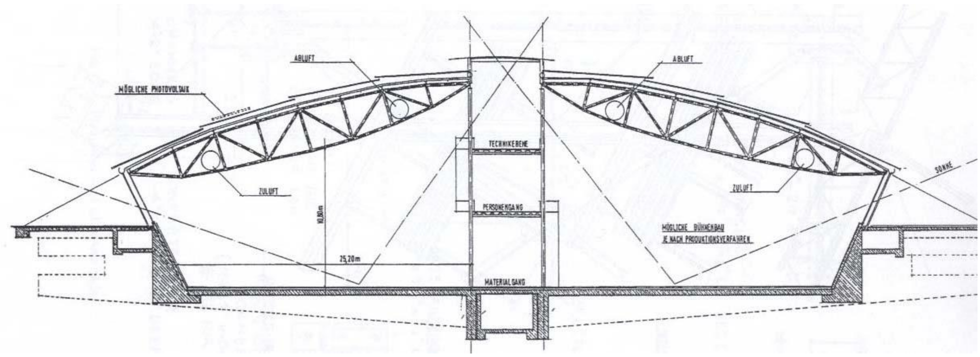


Fig: 1.15 Gerisha cooling system

Conclusion

Buildings mechanism of thermal behavior is in many details copy of a human body reaction. By his unconscious, instinctive behavior, with the defaulted characteristics, a man reduces or increases heat loss into the outer environment, through his blood circulation regulation toward the skin as a body's outer envelope. Besides, using their mind, humans with insulation which remains unchanged during both the summer and the winter season.

It is an advantage for a building, as opposite to a man a building uses cooling devices to reduce its temperature. However energy is necessary in such cases, which should be avoided in the present situation, regarding the energy crisis. perspiration, the only possible effect of a human body cooling in high temperature areas has not been used largely in buildings A few buildings around the world show it is possible option. Unfortunately, the impression is that building cooling was not the primary task of the water flow on facades, although it was one of the aims when designing those buildings. The task remains to expand it, but before each such building design, exact

calculations simulations, optimization should be done, resulting in total energy balance, taking into account water and energy balance.

The building envelopes are the main factor of building energy efficiency, as they represent a skin of a building's body. They should react as real skin as much as possible, defending interior from preheating, conserving it from heat losses in the heating season. That could be achieved relatively easy, but the architects and all factors influencing building design have to be working together in a manner of integral designing.

The energy needs analyze should be implemented in the building technical documentation, as a proof that the building will be a green one- at least from the point of view of energy efficiency and energy conservation. Computer programs that can be easily and quickly used on all locations have defined a meteorological year. For other locations, there are another methods based on degrees-days, or formulas given in some studies and recommended, as in Germany. The effect of water flowing above glass facades is in lowering the temperature of glass, and in reduced solar radiation transmittance.

ما قدمته من تطوير

بعد ما تقدم من موضوعات ومن تمارين حاولنا بها أن نغطي علم النقل الحراري في المباني نحب أن نختم هذا الجزء من الكتاب بعرض مثال حي على تطبيقات ومعالجات النقل الحراري في المباني.

تعودنا في جامعاتنا المختلفة أن نقول للطلاب إن العمارة الزجاجية لا تصلح لعالمنا العربي . وذلك بسبب إنتقال الحرارة الهائل من خلال الزجاج.

وكـان المعمـاريون إذا أرادوا أن وضـع مسـطح زجـاجي كعنصـر أسـاسـي وقـوي فـي الواجهـة اختـاروا الواجهـة الشـمالية، لأن الإشعاع الشمسـي لا يكـون مباشـراً فـي تلـك الواجهـة. علـى عكس ذلك الواجهة الجنوبية يحذر فتح أي فتحات ولو صغيرة بها. أما الواجهات الشرقية والغربية فهي أمر بين هذا وذاك.

كان هذا هو المتداول بين المعمارين وأساتذة العمـارة فـي مصـر والعالم العربي .

وفي عام 1992 قـام المهنـدس المعمـاري نيكـولاس جـريم شـو بتصميم مبنى Expo92 في أسبانيا ، ومنـاخ أسـبانيا قريـب مـن منـاخ مصـر حيـث أن الـدولتان مـن دول حـوض البحـر الأبـيض المتوسط.

لم يراعي جريم شو الكلام السابق وبنى علبة زجاجيـة أبعادهـا 20 x 60 x 16. علبـة زجاجيـة علـى الـرغم مـن إرتفـاع درجـة الحرارة!

فهل إعتمد على التكيف المركزي كما نصنع نحن؟

لا.. لقـد عـالج الزجـاج بطبقـة مـن المـاء البـارد بـين لـوحين مـن الزجاج ، وذلك من خلال دائـرة مغلقـة تقـوم بتبريـد المـاء وضـخه مرة أخرى إلى أعلى الواجهة.

قدم هذا الحـل فـرق فـي درجـات الحـرارة بـين الـداخل والخـارج مقـداره ثمانيـة درجـات. أي إذا كانـت درجـة الحـرارة الخارجيـة مقـدارها 40 درجـة مئويـة تكـون الحـرارة الداخليـة مقـدارها 32 درجة.

فقيل له إن درجة الأثنين وثلاثين غير كافيـة ، فهـي لا تقـع فـي منطقة الراحة الحرارية. فقال إذا سأقوم بتكرار هـذا الأمـر مـرتين على طريقـة House in House وأخصـص الجـزء الأبـرد ذو الأربعـة وعشرين درجة لكبار الزوار VIP . فقالوا لـه لقـد قسـمت البشـر إلى بشر مهم وبشر غير مهم . وذاك أمر ومخالف للإنسانية.

وأصبح هذا الأمر نقيصة في مشروع جريم شو . كنت في هذا الوقت أثناء تلك القضية الشائكة علميا والتي نزلت على صفحات الجرائد في أوروبا ، فأصبح الرأي العام طرفا فيها، كنت طالبا بالدراسات العليا – جامعة شتوتجارت .

لم يدخر أستاذي وسعا في تكليفي بتطوير مشروع جريم شو. على أن يكون التطوير يشمل معالجة هذا العيب الفادح في التصميم . وبنيت تصميمي على التالي

-معالجة السقف لا الواجهة بإعتبار السقف هو المعرض أكثر لأشعة الشمس (خمس أضعاف كمية الحرارة).

-تصميم التبريد بالمياة على أن تكون دائرة مفتوحة وليست مغلقة ، بحيث يتم التبريد بإحتكاك الهواء الخارجي بسطح المياة فتحدث عملية التبخر وبالتالي تزيد كفاءة التبريد بسبب طرد الحرارة بإسلوب البخر .

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-مراعاة وجود غطاء من الصاج على كامل السقف حتى لا يكون السطح المائي في حالة اتصال مباشر مع الشمس.

- جزء من المياة يتبخر والجزء الآخر يهبط إلى القاع ويتم تبريده طبيعيا عن طريق الحمل من خلال التبادل الحراري بين الأنابيب ورطوبة الأرض.

<u>النتائج التي تم التوصل إليها</u>

تم زيادة الفارق بين درجات الحرارة الداخلية والخارجية من 8 درجات إلى 17,3 درجة مئوية. هذه النتيجة يمكن تحسينها أيضا بزيادة سرعة جريان المياة.

رؤية مستقبلية

ليس الإبداع في البعد عن المشكلة –كما يحدث في جامعاتنا المصرية- حيث يمنعون الطالب من التعامل مع الواجهات الزجاجية خاصة الواجهة الجنوبية. لكن الإبداع في التعاطي مع المشكلة وحلها بأفضل الطرق.

ونحن إن كنا قدمنا حلا لتطوير حل نيكولاس جريم شو فهناك عدد لا نهائي من الحلول ينتظر المبدعين في أنحاء العالم.

إن المشاكل أمر محبب للمبدعين من المهندسين ، فهو لا وجود له بل لا كيان إلا بوجود المشكلة والقدرة على حلها ، وقد لمشروع Expo92 أبعد الأثر في نفسي. الأمر الذي أدى إلى إهتمامي بفيزياء المباني حيث إيجاد الحلول الموفرة للطاقة.

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Acoustics

Introduction

The fluid mechanics equations, from which the acoustics equations and results may be derived, are quite complicated. However, because most acoustics phenomena involve very small perturbations, it is possible to make significant significations to these fluid equations and to linearize them. The results are the equations of linear acoustics. The most important equation, the wave equation, is presented with some of its solutions. Such solutions give the sound pressure explicitly as functions of time and space, and the general may be termed the wave acoustics approach. This part of the book presents some of the useful results of this approach but also briefly discusses some of the other alternative approaches, sometimes termed ray acoustics and energy acoustics, which are used when the wave acoustics approach becomes too complicated.

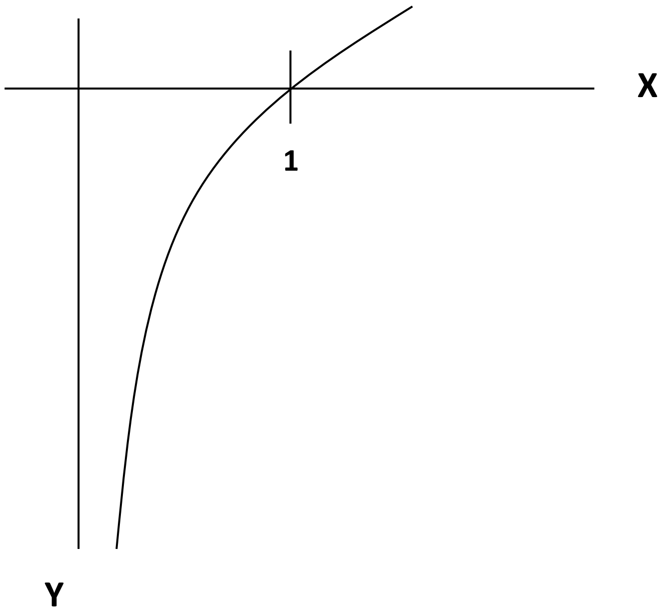
The first purpose of this chapter is to present some of the most important acoustics formulas and definitions, without derivation, which are used in the examples following. The second purpose is to make some helpful comments about the chapters that follow.

في الشق الأول من الكتاب بنينا ما استخلصنا من معادلات من علم النقل الحراري Heattransfer ، وفي هذا الجزءنبني على علم ديناميكا الموائع Fluidmechanics . ولأن إضطرابات الصوتية قليلة أو الحيز والفروقات التي نتحدث عنها في هذا الجزء من الكتاب قليلة أيضا يمكن لناإشتقاقها من معادلات ديناميكا الموائع ، التي هي بطبيعتها معادلات لوغاريتمية .

ومن خصائص تلك المعادلات فضلا عن الفروقات الضئيلة التي نحصل عليها كون تناسبها الطردي غير ثابت . فإذا كانت شدة الصوت لمتحدث يبعد عنك ثلاثة أمتار مقدارها ستون ديسبل ، لن تكون ثلاثون إذا ما تضاعفت المسافة.

ملاحظة أخرى ينبغي الإشارة إليها قبل البدء في هذا الجزء من الكتاب. أنه لا يوجد حقيقة صفر ديسبل ، فالمعادلة اللوغاريتمية من خصائصها أنها لا تتقابل مع محور x حتى في اللا نهاية.

$$e^x = \frac{1}{ln}$$



وعليه فإن مقولة لا يوجد في الطبيعة صفر ديسبل مقولة حقيقة. ويجدر الإشارة أيضا أن المجال السمعي للإنسان محدود بين 16 و 20000 هيرتز. شأنه في ذلك شأن جميع حواس الإنسان ، فالإنسان محدود في سمعه ، محدود في بصره ، محدود في عقله.

Linear Acoustic

Acoustics

Acoustics is the science of sound. We interpret sound through our sense of hearing. Anything that is interpreted by the senses is open to subjectivity in terms of likes and dislikes. This subjective interpretation of sound not only defines the difference between music and noise, but also dictates the quality of communication within a space. People often think of acoustics as a narrow, esoteric field that has little practical application short of designing concert halls. But the field has many practical branches, including noise control, psychoacoustics (the psychological effects of sound on people), physiological acoustics (the physical effects of sound on people) and bioacoustics (the use of sound waves in medical diagnostics.

إن هذا العلم الذي نحن بصدده يختص بالأصوات ونشعر به من خلال الحاسة السمعية. وأي شيئ يدرك بالحواس معرض إلى كلمتي أرغب ولا أرغب ، وأحب وأكره ، وعلى الرغم من هذا الكلام الذي لا يخضع لقواعد علمية ثابتة إلا أننا نفرق بين التخاطب والموسيقى أيضا بحاسة السمع ، ونفرق بين الهدوء والإزعاج أيضا بحاسة السمع . ومن ثم فهو أداة القياس المتداولة بين الناس.

Sound Generation

Sound is generated when pressure-oscillations are generated in an elastic medium at rates that are detectable by a hearing mechanism. For simplicity and practicality, the sound waves described are in the range detectable by most humans.

Sound generation is a physical phenomenon while noise is a subjective interpretation of sound. Therefore, if a tree falls in the

woods and no one is nearby to hear it, the tree has generated a sound but not noise (putting to rest a long-debated argument). If the source of pressure oscillation is stationary (relative to a stationary observer) and physically small compared to your distance away from that source, we call it point source.

ينشأ الصوت بإختلاف ذبذبات الضغط داخل وسط مرن. وإذا أردنا الإفصاح أكثر نقول إن فراغ الفم يوجد به ضغط ما قبل حركة اللسان فإذا ما تحرك اللسان نشأ ضغط جديد وبذا يفصح الإنسان عما بداخله.

إذا فالموجة الصوتية موجة ميكانيكية والموجة الحرارية القادمة من الشمس موجة كهرومغناطيسية والفارق بين الموجتان أن ال Vacuum وهو الفراغ الموجود بين لوحين من الزجاج المفرغ يعزل الموجة الميكانيكية فلا يسمح لها بالمرور ولا يعزل الموجة الكهرومغناطيسية. أي أن الشمس تعبر من خلال نافذة الطائرة ولا يعبر صوت المحركات. ولابد من القول أنه كلما زاد سمك الشباك كلما زادت القدرة على العزل

If this source is oscillating at a constant rate, it generates a pure tone and the source can be described in terms of a single frequency, or rate of oscillation. This frequency is usually described in terms of units o cycles (of oscillations) per second, also labeled as hertz (abbreviated Hz), named after the German physicist Heinrich Hertz, who is credited with discovering electromagnetic radiation waves.

The constant rate oscillation with the passage of time translates to the sine wave. One complete cycle of oscillation is shown ending at the point at which the sine wave begins to repeat its pattern. Pure tones rarely exist in nature since most sound comprises contributions from many audible frequencies.

It is generally accepted that humans can hear frequencies between 20 and 20000 Hz. Within this frequency range, we are most sensitive to sounds having frequency components between 500 and 4000 Hz. This, by no coincidence, also corresponds to the dominant frequency range generated by the human voice. Although most or us can still detect low pitches between 20 and 500Hz and high pitches between 4000 arid 20000 Hz, our hearing mechanisms are less sensitive to these sounds.

Frequency(Hz)	**Wavelength**
20	56ft/17m
50	23ft/7m
100	11ft/3m
500	2ft/0.7m
1000	1ft/0.3m
5000	0.2ft/0.07m
10000	0.1ft/0.03m
20000	0.06ft/0.02m

نلاحظ العلاقة بين التردد والطول المـوجي فعنـد 20 هيرتـز يكـون الطـول الموجي 17متر وعند 20 ألف هيرتز تكـون قيمـة الطـول المـوجي 2سـم. ولابـد مـن الإشـارة أنـه هنـاك العديـد مـن الكائنـات التـي تتمتـع بمـدى سـمعي أكبر من الإنسـان مثل الخفاش والكلاب.. وهناك أيضا العديد من الإسـتخدامات الطبية للموجات فـوق الصـوتية كتفتيـت الحصـوة وغيرهـا.

ومن ثم فزيادة التردد ونقصان الطول الموجي يؤدي إلى التحكم في الموجة ثم استخدامها في الأغراض المختلفة أما العكس فهي طاقة مبعثرة لايمكن الإستفادة منها.

Waves motions

Some of the basic concepts of acoustics and sound wave propagation are discussed here. Wave motion is easily observed in the waves on stretched strings and as ripples on the surface of water. Waves on strings and surface water waves are very similar to sound waves in air (which we cannot see), but there are some differences that are useful to discuss. If we throw a stone into a calm lake, we observe that the water waves (ripples) travel out from the point where the stone enters the water. The ripples spread out circularly from the source at the wave speed which is independent of the wave height. Somewhat like the water ripples, sound waves in air travel at a constant speed, almost independent of air propagate by transferring momentum and energy between air particles. There is no net flow of air away from a source of sound, just as there is no net flow of water away from the source of water waves. Of course , the waves on the surface of a lake are circular or two dimensional, while in air, sound waves in general are spherical or three dimensional.

As water waves move away from a source, their curvature decreases, and the wave fronts may be regarded almost as straight lines. Such waves are observed in practice as breakers on the seashore. A similar situation occurs with sound waves in the atmosphere. At large distances from a source of sound, the spherical wave front curvature decreases, and the wave fronts may be regarded almost as plane surfaces.

Plane sound waves may be defined as waves that have the same acoustic properties at any position on a plane surface drawn perpendicular to the direction of propagation of the wave. Such plane sound waves can exist and propagate along straight tube or duct (such as an air-conditioning duct). In such a case, the waves propagate in a direction along the duct and the plane waves are perpendicular to this direction (and are represented by ducts cross sections). Such waves in a duct are one dimensional, like the waves on along string or rope under tension (or like the ocean breakers described above).

إذا قمنا بإلقاء حجر في بحيرة هادئة فإننا نلمح إنتقالا للموجات من مصدر الإلقاء بسرعة منتظمة وشكل دائري ، وذلك شأن الموجات الصوتية تسافر في الهواء من خلال مصدر للصوت لكنها إنتشارها في الهواء يكون بشكل حلزوني وبالتالي ثلاثي الأبعاد أما الإنتشار الأول في البحيرة الهادئة فهو ثنائي الأبعاد.

Although there are many similarities between one-dimensional sound waves in air, waves on strings, and surface water waves, there are some minor differences. In a fluid such as air, the fluid particles vibrate back and forth in the same direction as the direction of wave propagation; such waves are known as a longitudinal, compressional, or sound waves. On a stretched string, the particles vibrate at right angles to the direction of wave propagation; such waves are usually known as transverse waves.

The surface water waves described are also partly transverse partly longitudinal waves, with the complication that the water particles move up and down and back and forth horizontally. (This movement describes elliptical paths in shallow water and circular paths in deep water. The vertical particles motion is much greater than the

horizontal motion for shallow water, but the two motions are equal for deep water.) The water wave direction is, of course, horizontal.

Surface water waves are not compressional (like sound waves) and are normally termed surface gravity waves. Unlike sound waves, where the wave speeds is independent of frequency, long wavelength waves, and thus water wave motion is said to be dispersive. Bending waves on beams, plates, cylinders, and others engineering structure are also dispersive. There are several other types of waves that can be of interest in acoustics: shear waves, torsional waves, and boundary waves, but the discussion here will concentrate on sound wave propagation in fluids.

إن أوجه الشبه والقصور بين الموجات الصوتية وموجات المياة كبيرة لكنها إلى أوجه الشبه أقرب فالموجتان تنتشران بإتجاه واحد وإن كان سطح الماء لا يضغط على الموجه فيى إتجاهها العكسي كما هو الحال في الموجة الصوتية

Plane sound waves

If a disturbance in a thin element of fluid in a duct is considered, a mathematical description of the motion may be obtained by assuming that:

1) The amount of fluid in the element is conserved
2) The net longitudinal force is balanced by the inertia of the fluid in the element.
3) The process in the element is adiabatic (there is no flow of heat in or out of the element)
4) The undisturbed fluid is stationary (there is no fluid flow)

Then the following equation of motion may be derived.

$$\frac{d^2p}{dx^2} - \frac{d^2p}{dxt^2}\frac{1}{c^2} = 0 \qquad (2.1)$$

This equation is known as the one-dimensional equation of motion, or acoustic wave equation, and it relates the second rate of change of the sound pressure with the coordinate x with the second rate of change of the sound pressure with time t through the square of speed of sound c. Identical wave equations may be written if the sound pressure p in Eq(1) is replaced with the particle displacement ε, the particle velocity u fluctuating density ρ, or the fluctuating temperature T. However, the wave equation in terms of the sound pressure in eq(1) is perhaps most useful, since the sound pressure is the easiest acoustic quantity to measure (using a microphone) and is the acoustic perturbation we sense with our ears in the sound pressure P is the acoustic pressure perturbation or fluctuation about the time-averaged, or undisturbed, pressure P0.

The speed of sound waves c is given for a perfect gas by:

$$C = (\gamma.R.T)^{\frac{1}{2}} \qquad (2.2)$$

The speed of sound is proportional to the square root of the absolute temperature T. The ratio of specific heats γ and the gas the gas constant R are constant for any particular gas. Thus Eq(2) may be written as

$$C = C0 + 0.6\ Tc \qquad (2.3)$$

Where, for air, C_0 = 331.6m/s, the speed of sound at 0°c, and T_c is the temperature in degrees Celsius. Note that Eq.(2.3) is an approximate formula valid for T_c near room temperature.

ومن هذا نعلم أن الصوت تتغير سرعته بتغير الحرارة وذلك من المعادلة الثانية والثالثة أو بتغير X أو T من المعادلة الأولى.

وعلى كل الأحوال فسرعة الصوت عند درجة حرارة صفر 331.6m/s وتزيد السرعة بزيادة درجة الحرارة. غير أنه ينبغي أن يكون واضحا للجميع أن لا مقارنة بين سرعة الصوت وسرعة الضوء التي تبلغ 300.000Km/s .

ولكن ما العلة في أن سرعة الصوت تزيد بزيادة درجة الحرارة؟

هذا السؤال تجيب عنه معادلة المنحنى الإفتراضي 2.4 حيث أن الدالة المتعلقة بX المسافة و T الزمن ليست معاجدلة سينية صريحة ومن ثم فإن الإحتكاك بين الموجة الصوتية والفوتونات هو الذي يرسم هذا المنحنى، فإذا ما زادت درجة الحرارة قلت الكثافة بين الجزيئات وبالتالي زادت السرعة .

The speed of sound in air does not depend on the atmospheric pressure. A solution to (1) is

$$P = f_1(ct-x) + f_2(ct+x) \qquad (2.4)$$

Where f_1 and f_2 are arbitrary function such as sine, cosine exponential, log, and so on. It is easy to show that Eq.(4) is a solution to the wave equation (1) by differentiation and substitution into Eq.(2.1). Varying x and t in Eq(2.4) demonstrates that $f_1(ct-x)$ represents a wave traveling in the positive x-direction with wave

speed c, while f2(ct+x) represents a wave traveling in the negative x-direction with the wave speed c, (see Fig2.1)

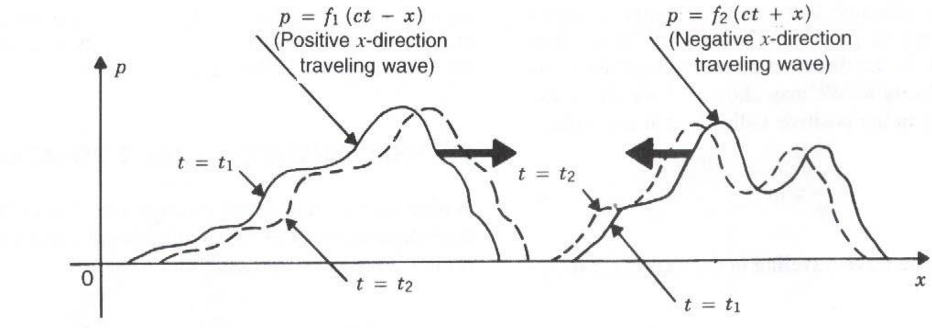


Fig.2.1 Plane waves of arbitrary wave form [1]

The solution given in eq(2.4) is usually known as the general solution, since, in principle, any type of sound wave from is possible. In practice, sound waves are usually classified as impulsive or steady in time. One particular case of a steady wave is of considerable importance. Waves created by sources vibrating sinusoid-ally in time (e.g., a loudspeaker, a piston, or a more complicated structure vibrating with a discrete angular frequency ω) vary both in time t and space x in a sinusoidal manner (see fig.2.2)

$$P = p1\sin(\omega t-kx+\theta 1) + p2\sin(\omega t+kx+\theta 2) \qquad (2.5)$$

لكن لابد لنا أن نفرق بين الشكل العام للموجة في معادلة 2.4 والشكل الخاص المبسط في معادلة 2.5 وهي معادلة سينية صريحة . مع العلم أن P1 تسافر في الإتجاه الإيجابي وP2 في الإتجاه السلبي.

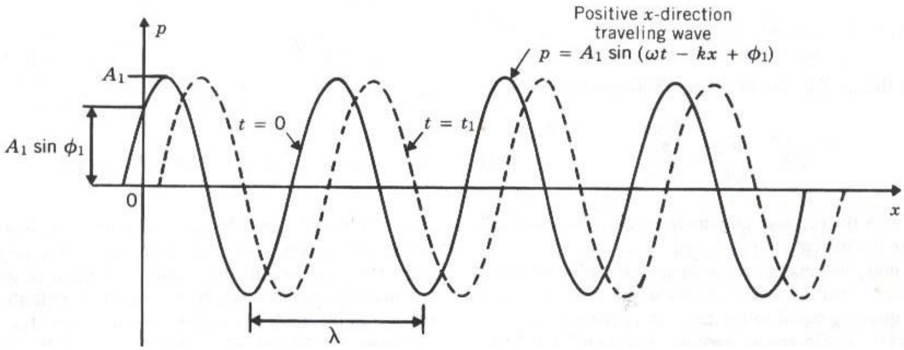


Fig:2.2 Simple harmonic plane waves [1]

At any point in space, x, the sound pressure p is simple harmonic in time. The first expression on the right of eq.(2.5) represents a wave of amplitude p1 traveling in the positive x-direction with speed c, while the second expression represents a wave of amplitude p2 traveling in the negative x-direction. The symbols θ1 and θ2 are phase angles, and k is the acoustic wave-number. It is observed that the wave-number k=ω/c by studying the ratio of x and t in Eqs. (2.4) and (2.5). At some instant t the sound pressure pattern is sinusolidal in space, and it repeats itself each time kx in increased by 2π. Such a repetition is called a wavelength λ. Hence, kλ=2π or k=2π/λ. This gives ω/c= 2πf/c =2π/λ

$$\lambda = \frac{c}{f} \qquad (2.6)$$

The wavelength of sound becomes smaller as the frequency is increased. At 100Hz, λ=3.5m= 10ft . At 1000 Hz, λ=0.35m = 1 ft

At some point x in space, the sound pressure is sinusoidal in time and goes through one complete cycle when ω increases by 2π. The time for a cycle is called the period T. Thus, ωT=2π, T=2π/ω, and

$$T = \frac{1}{f} \qquad (2.7)$$

وبمناسبة الحديث عن سعة الموجة أحب أن أشير أن سعة الموجة الكبيرة لا تخدم تطبيقات الموجات الصوتية. فالموجات فوق الصوتية ultrasonic ذات تردد عالي وسعة موجية صغيرة.

وبشكل عام تطبيقات الموجات الصوتية تكون من خلال حزمة يمكن التحكم فيها وتوجيهها ،أما السعة الكبيرة فتضعف الموجة ولا تجعلها قابلة لللإستخدام .

Impedance and sound intensity

We see that for the one-dimensional propagation considered the sound wave ظ travel with a constant wave speed c, although there is no net, time-averaged movement of the air particles. The air particles oscillate back and forth in the direction of wave propagation (x-axis) with velocity u. We may show that for any plane wave traveling in the positive x-direction at any instant

$$\frac{p}{u} = \rho c \qquad (2.8)$$

And for any plane wave traveling in the negative x-direction

$$\frac{u}{p} = -\rho c \qquad (2.9)$$

The quantity ρc is called the characteristic impedance of the fluid, and for air, ρc = 428kg/m².s at 0°c and 415kg/m².s at 20°c.

كما سبق وأن أشرنا إلى أن جزيئات الهواء "الفوتونات" تؤثر بسرعتها على الموجة الصوتية ، سواء من حيث السرعة أو من حيث الكثافة. وبالتالي فمعادلة 2.8 يفهم منها أن الضغط على سرعة الفوتونات يساوى الكثافة في سرعة الموجة الصوتية. إذا فيمكننا حساب سرعة الموجة الصوتية بطريقة أخرى عنم التي تعرفنا عليها من قبل "راجع معادلة 2.6 "

The sound intensity is the rate at which the sound wave dose work on an imaginary surface of unit area in a direction perpendicular to the surface. Thus, it can be shown that the instantaneous sound intensity in the x-direction, I, is obtained by multiplying in the instantaneous sound pressure p by the instantaneous particle velocity in the x-direction, u. therefore

$$I = p.u \qquad (2.10)$$

And for a plane wave traveling in the positive x-direction this becomes

$$I = \frac{p^2}{\rho c} \qquad (2.11)$$

The time-averaged sound intensity for a plane wave traveling in the positive x-direction, (I), is given as

$$(I)t = \frac{(p^2)t}{\rho c} \qquad (2.12)$$

And for the special case of a sinusoidal (pure-tone) wave

$$(I)t = \frac{(p^2)t}{\rho c} = \frac{p^2\ 1}{2\ \rho c} \qquad (2.13)$$

Where P1 is the pressure amplitude, and the mean-square pressure is thus $(p^2)t = p^2ms = \frac{1}{2}p^21$. We note, in general, for sound propagation in three dimensions, that the instantaneous sound intensity I is a vector quantity equal to the product of the sound pressure and the instantaneous particle velocity u. Thus I has magnitude and direction. The vector intensity I may be resolved into components Ix, Iy and I.

Three-dimensional wave equation

In most sound fields, sound propagation occurs in two or three dimensions. The three-dimensional version of Eq.(1) in Cartesian coordinates is

$$\frac{d^2p}{dx^2} + \frac{d^2p}{dy^2} + \frac{d^2p}{dz^2} - \frac{1}{c^2}\ \frac{d^2p}{dt^2} = 0 \qquad (2.14)$$

This equation is useful if sound wave propagation in rectangular spaces such as rooms is being considered. However, it is helpful to recast Eq.(2.14) in spherical coordinates if sound propagation from source of sound in free space being considered. It is a simple mathematical procedure to transform Eq.(2.14) into spherical coordinates, although the resulting equation is quite complicated.

إذا كان إنتشار الصوت بشكل حلزوني وكان الفراغ الذي ينتشر فيه مستطيل يكون هذا الإنتشار ثلاثي الأبعاد ويطبق فيه معادلة 2.14

However, for propagation of sound waves from a spherically symmetric source (such as the idealized case of a pulsating spherical balloon known as a unidirectional or monopole source), the equation becomes quite simple (since three is no angular dependence):

$$\frac{d^2(rp)}{dr^2} - \frac{1}{c^2}\frac{d^2(rp)}{dt^2} = 0 \qquad (2.15)$$

Here, r is the distance from the origin and p is the sound pressure at that distance. Equation (2.15)is identical in form to equation 1with p replaced by rp and x by r. The general and simple harmonic solutions to eq.(2.15) are thus the same as Eqs. (2.4) and (2.5) with p replaced by rp and x with r. t

$$rp = f1(ct-r) + f2(ct+r) \qquad (2.16)$$

or

$$p = \frac{1}{r} f1(ct-r) + \frac{1}{r} f2(ct+r) \qquad (2.17)$$

where f1 and f2 are arbitrary functions. The first term on the right of Eq.(2.17) represents a wave travelling outward from the origin; the sound pressure p is seen to be inversely proportional to the distance r. If the distance r is doubled, the sound pressure level [Eq.(2.29)] decreases by 20log10(2) = 20(0.301) = 6dB. This is known as the inverse-square law.

The second term in Eq(2.17)represent sound wave traveling inward toward the origin, and in most practical cases these can be ignored (if reflecting surface are absent).

The simple harmonic (pure-tone) solution of Eq.(2.15) is

$$P = \frac{p1}{r}\sin(\omega t - kr + \theta 1) + \frac{p2}{r}\sin(\omega t - kr + \theta 2) \qquad (2.18)$$

Where p1 and p2 are the sound pressure amplitudes.

في المعادلة رقم 2.18 نرى بوضوح أن الموجة الصوتية المنطلقة من مصدر ما تتناسب قوتها عكسيا مع المسافة r فإذا ما ضاعفنا المسافة فإننا نجد أن القوة قلت بمقدار 12dB .

ونحب أن ننوه ونحن ضمن ثنايا هذا الكتاب خاصة ونحن نتكلم عن الصوت ، أن هناك العديد من المباني التي تقع على مقربة من مصادر شديدة الإزعاج كشريط السكة الحديد ومهبط الطائرات. ومن ثم فإن تناول مثل هذه الموضوعات ليس أمرا ترفيا ، بل أنه يصل إلى حد الإلزام والوجوب عند تصميم المسارح وقاعات المحاضرات العامة.

هذا فضلا عن أن الموجات الصوتية لها القدرة على إزالة مبنى وقد حدث.

Sources of sound

The second term on the right of Eq.(2.18), as before, represents sound waves traveling inward to the origin and is of little practical interest. However, the first term represents simple harmonic waves of angular frequency ω traveling outward from the origin, and this may be rewritten as :

$$P = \frac{\rho\, c\, k\, Q}{4\, \pi\, r}\sin(\omega t - kr + \theta 1) \qquad (2.19)$$

Where Q is termed the strength of an omnidirectional (monopole) source situated at the origin, and $Q=4\pi p1/\rho ck$. The mean-square pressure may be found by time-averaging the square of Eq.(19) over a period T:

$$P^2 rms = \frac{(\rho ck)^2 Q^2}{32\pi^2 r^2} \qquad (2.20)$$

From Eq.(2.20), the mean-square pressure is seen to vary with the inverse square of the distance r from the origin of the source for such an idealized omnidirectional point source everywhere in the sound field. Again, this is known as the inverse-square law. If the source is idealized as a sphere of radius a pulsating with a simple harmonic velocity amplitude U, we may show that Q has units of volume flow rate (cubic meters per second). If the source radius is small in wavelengths so that a < λ or ka < 1, then we can show that the strength $Q=4\pi a^2 U$.

Many sources of sound are not like the simple omnidirectional monoplole source just described. For example, an unbaffled loudspeaker produces sound both from the back and front of the loudspeaker. The sound from the front and the back can be considered as two sources that are 180° out of phase with each other. This system can be modeled as two out-of-phase monopoles of source strength Q separated by a distance l. The sound pressure produced by such a dipole system is

$$P = \frac{\rho ckQl \cos\theta}{4\pi r} \cdot \left[\frac{1}{r}\sin(\omega t - kr + \theta) + \quad k\cos(\omega t - kr + \theta)\right] \qquad (2.21)$$

Where θ is the angle measured from the axis joining the two sources (the loudspeaker axis in the practical case). Unlike the monopole, the dipole field is not omnidirectional. The sound pressure field is directional. It is, however, symmetric and shaped like a figure-eight with its lobes on the dipole axis.

For a dipole source of the sound pressure has a near-field and a far field behavior similar to the particle velocity of a monopole. Close to the source (the near field), for some fixed angle θ, the sound pressure falls off rapidly, while far from the source (the far field kr>1), the pressure falls off more slowly. In the near field the sound pressure level decreases by 12 dB for each doubling distance r.

In the far field the decrease in sound pressure level is only 6dB for doubling of r like a monopole. The phase of sound pressure also change with distance r, since close to the source the sine term dominates and far from the source the cosine term dominates. The particle velocity may be obtained from the sound pressure [Eq(2.21)] and use of Eulers equation [Eq.(2.22)]. It has an even more complicated behavior with distance r than the sound pressure, having three distinct regions.

Sound intensity and directivity

The radial particle velocity in a spherically spreading sound field is given by Euler s equation as

$$U = -\frac{1}{\rho} \cdot \int \frac{dp}{dr}\, dt \qquad (2.22)$$

and substituting Eqs. (2.19) and (2.22) into (2.10) then using Eq.(2.20)_and time-averaging gives the magnitude of the radial intensity in such a field as

$$I = \frac{p^2}{\rho c} \qquad (2.23)$$

The same result as for a plane wave [see Eq.(2.12)]. The sound intensity decreases with the inverse square of the distance r. Simple omnidirectional monopole sources radiate equally well in all directions. More complicated idealized source such as dipoles, quadruploles, and vibrating piston sources create sound fields that are directional (see Fig.2.3)

ينبغي على المصمم أن يرعي ما إذا كان مصدر الصوت مصدر أحادي أم مصدر متعدد. وعلى سبيل المثال لا الحصر فإن مضاعفة المسافة في الحالة الأولى يؤدي إلى نقصان شدة الصوت بمقدار 6dB ، أما في الحالة الثانية فيؤدي إلى النقصان بمقدار 12dB .

وليس هذا فحسب ، بل إن طبيعة مصدر الصوت قد تؤدي إلى استخدام بعض المعادلات دون الأخرى وذلك بناء على المعطيات في الحالة التي نحن بصددها فإذا كان مصدر الصوت ماكينة أو ألة يكون قياس كثافة الصوت أكثر تعقيداً.

Of course, real source such as machines produce even more complicated sound fields than these idealized sources. (For a more complete

Discussion of the sound fields created by idealized sources) However, the same result as Eq.(2.23) is found to be true for any source of sound as long as the measurements are made sufficiently far from the source. The intensity is not given by the simple result of Eq.(2.23) close to source such as dipoles, quadrupoles, or more complicated source of sound close such sources Eq.(2.10) must be used for the instantaneous radial intensity, or

$$I_t = (p.u)t \qquad (2.24)$$

For the time-averaged radial intensity

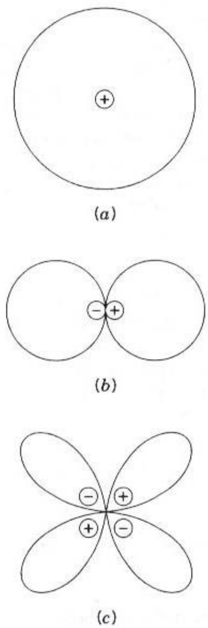


Fig.2.3 Polar directivity plots for the radial sound intensity in the far field of (a) monopole, (b) dipole, and (c) lateral quadruple.[2]

The time-averaged radial sound intensity for a dipole is given by

$$I = \frac{\rho c k^4 (Ql)^2 cos^2\theta}{32\ \pi^2 r^2} \qquad (2.25)$$

The sound intensity radiated by a dipole is seen to depend on $cos^2\theta$. In general, a directivity factor $D_{\theta,\Phi}$ may be defined as the ratio of the radial intensity $(I_{\theta,\Phi})_t$ (at angle θ and Φ and distance r from the source) to the radial intensity $(I_s)_t$ at the same distance r from an omnidirectional source of the same total power. Thus

$$D_{\Phi,\theta} = \frac{I_{\Phi,\theta}}{I_s} \qquad (2.26)$$

A directivity index DI $_{\Phi,\theta}$ may be defined, where

$$D_{\Phi,\theta} = 10\log D_{\Phi,\theta} \qquad (2.27)$$

Sound power

The sound power P of a source is given by integrating the intensity over any closed surface S around the source (see Fig.2.4)

$$P = \int_s (In)t\, ds \qquad (2.28)$$

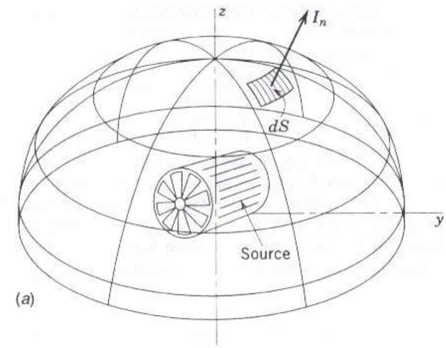


Fig.2.4 Normal component of sound intensity In being measured on a segment ds of a hemispherical enclosure surface [2]

The normal component of the intensity In must be measured in a direction perpendicular to the elemental area ds. If the spherical surface is chosen, then the sound power of an omnidirectional (monopole) source is

$$P_m = (I_r)_t \cdot 4\pi r^2 \qquad (2.29)$$

$$P_m = \frac{p^2}{\rho c} \cdot 4\pi r^2 \qquad (2.30)$$

And from Eq.(2.20) the sound power of a monopole is

$$P_m = \frac{\rho c k^2 Q^2}{8\pi} \qquad (2.31)$$

It is apparent from Eq.(31) that the sound power of an idealized (monopole) source is independent of the distance r from the origin, where the source is located. This is the result required by conservation of energy and also to be expected for all sound sources.

وقياس قوة الصوت يكون بتكامل الكثافة على سطح مقعر وعمودي على مركز الصوت. ولاشك أن بعد المصدر عن المستقبل يؤثر على شدة الصوت . إلا أننا كنا قد ألمحنا ونبهنا إلى أن التناسب العاكسي في هذه الحالة ليس تناسبا متكافأً ، بمعنى إذا كان شدة الصوت على بعد واحد متر خمسين ديسبل ، فهي لن تكون بحال من الأحوال خمسة وعشرين إذا أضحت المسافة مترين. وذاك طبيعة المعادلة اللوغاريتمية.

Equation (30) shows that for an omnidirectional source (in the absence of reflections) the sound power can be determined from measurements of the mean square pressure made with a single microphone. Of course, for such a source, measurements should

really be made with a reflection-free (anechoic) environment or very close to the source where reflection are presumably less important.

The sound power of a dipole source is obtained by integrating the intensity given by eq.(.225) over a sphere around the source. The result for the sound power is

$$P_d = \frac{\rho c k^4 Q l^2}{24\pi} \qquad (2.32)$$

The dipole is obviously a much less efficient radiator than a monopole, particularly at low frequency. In practical situations with real directional sound source and where reflection are important, use of Eq.(30) becomes difficult and less accurate, and then the sound power is more conveniently determined from eq.(28) with a sound intensity measurement system.

ويستغرب القارئ الكريم حين نقول أن فاعلية شدة الصوت في المصدر الأحادي أكبر بكثير من فاعليته في المصدر المتعدد. وعلى وجه الأخص في الترددات المنخفضة.

معنى هذا هل يختلف الوضع بإختلاف نوعية التردد؟

الإجابة : بالتأكيد نعم.

It is important to note that the sound power radiated by a source can be significantly affected by its environment. For example, if a monopole source with a high internal impedance (whose strength Q will be unaffected by the environment) is placed on a floor, its sound power will be double (and its sound power level increased by 3dB). If it is placed at a floor-wall intersection, its sound power will be increased by four times (6dB); and if it is placed in a room corner, its power is increased by eight times (9dB).

Decibels and levels

The range of sound pressure magnitude and sound power experienced in practice is very large (see Figs. 2.5 and 2.6). Thus, logarithmic rather than linear measures are often used for sound pressure and power. The most common is the decibel. The decibel represents a relative measurement or ratio. Each quantity in decibels is expressed as a ratio relative to a reference sound pressure, power, or intensity. Whenever a quantity is expressed in decibels, the result is known as a level.

The decibel (dB) is the ratio R1 given by

$$\text{Log10R} = 0.1, \quad \text{Log10R} = 1\text{dB} \qquad (2.33)$$

Thus, R1= $10^{0.1}$ = 1.26 . The decibel is seen to represent the ratio 1.26. A larger ratio, the bel is sometimes used.

The bel is the ratio R2 given by log10R2=1. Thus, R2= 10^{1} = 10. The bel represents the ratio 10.

The sound pressure level L_p is given by

$$L_p = 10\log10\left(\frac{p^2}{pr^2}\right) \text{ which } P_r = \text{P reference} \qquad (34)$$

$$= 20 \log10 \frac{p}{pr}$$

Where P_{ref} is the reference pressure P_{ref} =20µ Pa= 0.00002 N/m² = 0.0002 µ bar. This reference pressure was originally chosen tp correspond to the quietest sound (at 1000Hz) that the average young person can hear.

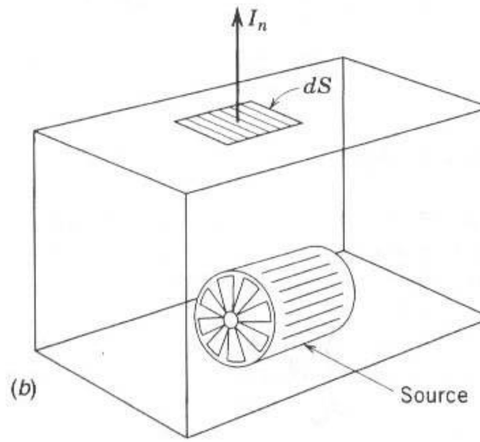


Fig:2.5 Some typical sound pressure levels [2]

بالإضافة إلى ما قلناه عن طبيعة المعادلة اللوغاريتمية فإن وحدة قياس الديسبل هي وحدة نسبية ratio وعليه فلا يوجد صفر ديسبل. ومستوى الصوت يقاس بتقسيم الضغط الحادث على الضغط القديم.

في شكل 2.5 يوجد ضغط داخل الفراغ قبل تشغيل الموتور ، وبع التشغيل يوجد ضغط جديد ، ومستوى الصوت الكائن في هذا الفراغ هو من تقسيم الضغط الجديد على الضغط القديم.

يحدث هذا الشيئ داخل تجويف الفم. فالله الذي أبدع الإنسان وخلقه في أحسن تقويم ، خلق له لسان داخل غرفة هي تجويف الفم، وحركة اللسان داخل هذا التجويف تنشئ ضغطين. والموجة الميكانيكية الناشئة تخرج في صورة حديث أو غناء أو مشاجرة.

إنه الإبداع في تصميم الإنسان .. وكيف والذي أبدعه هو من يقول للشيء كن فيكون.

بقي أمر ما ، كنتى كثيراً ما أقول لطلابي فـي قاعـة المحاضـرة، خاصـة طلاب قسـم الهندسـة المعماريـة ، إن أي موجـة صـوتية داخـل فـراغ مـا يحدث لها أحد ثلاث أمور

- إنتقال من فراغ إلى آخر
- إنعكاس للموجة الصوتية
- إمتصاص للموجة الصوتية

وكنت أتسأل أين يذهب الجزء الممتص؟

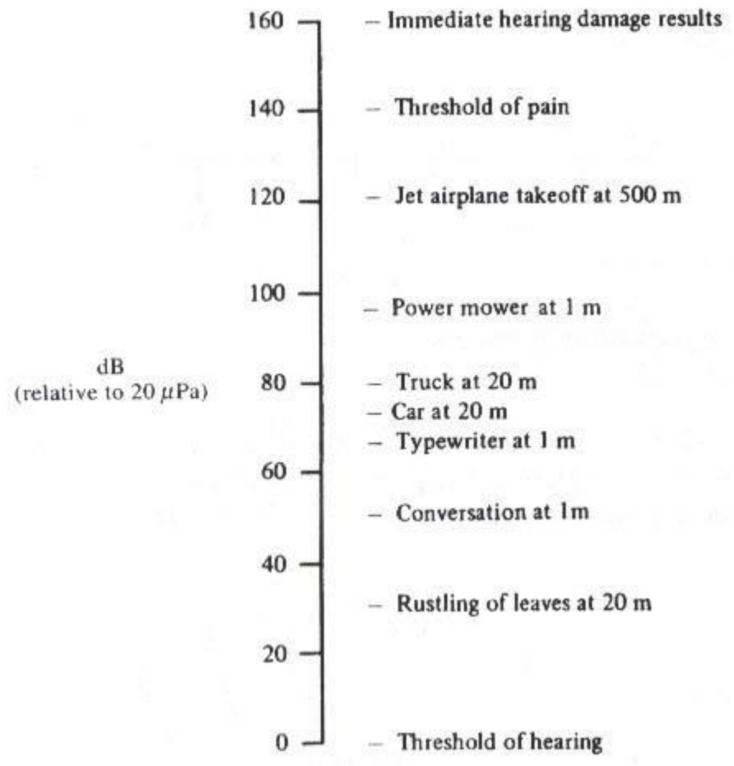


Fig:2.6 Some typical sound power levels, L_p [3]

كـان الطـلاب يصـمتون عنـد هـذا السـؤال ..بـاهتون ..حـائرون.. لاتتحـرك ألسـنتهم. فكنـت أقـول لهـم درسـتم قـديما فـي قـوانين نيـوتن الطاقـة لاتفنى ولا تستحدث من العدم. والجزء الممتص من الصوت يخـرج علـى شـكل حرارة ولكنك لا تشعر بها لصغر حجم مقدار الحرارة الناشـئة.

إذا فالموجـــة الميكانيكية"الصـــوت" تغيـــر مـــن شـــكلها إلـــى موجـــة كهرومغناطيسـية "حرارة" وبالتالي فالصوت طاقة.

وفي الشـكل 2.6 ، 2.7 يظهر هذا بوضـوح ونسـتطيع أن نقـرأ أن كـم مـن الديسـبل يعادل كم من الوات .

The sound power level L_p is given by

$$L_p = 10\log 10 \frac{p}{Pref} \qquad (2.35)$$

Where P is the sound power of a source and Pref= 10^{-12} W is the reference sound power.

The sound intensity level L_i is given by

$$L_i = 10\log 10 \frac{I}{Iref} \qquad (2.36)$$

Where I is the component of the sound intensity in a given direction and Iref = 10^{-12}W/m² is the reference sound intensity. Some typical sound pressure and power levels are given in Fig:2.5 and 6. If two sound source radiate independently (uncorrelated source), then because the interfering effects of the source pressure time-average to zero, the mean square pressure are additive and the total sound

pressure level at some point in space, or the total sound power level, may be determined using fig:2.7

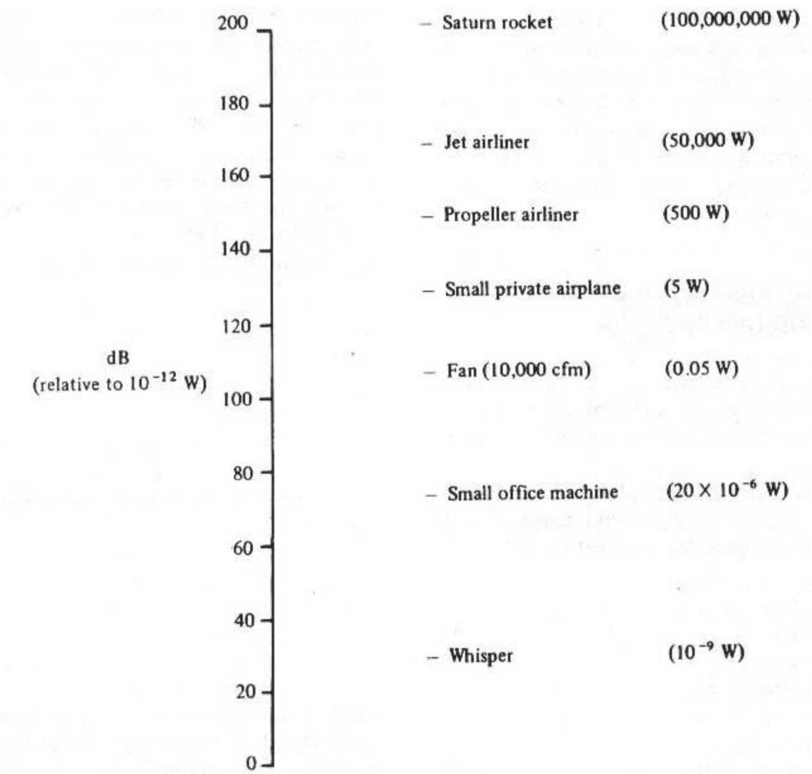


Fig:2.7 Chart for combination levels of decibels (for uncorrelated sources)

Reflection, scattering and diffraction

For a homogeneous plane sound wave at normal incidence on a fluid medium of different characteristic impedance ρc, both reflected and transmitted waves are formed (see Fig:2.8)

From energy considerations (provided no losses occur at the boundary) the sun of the reflected intensity Ir and transmitted intensity It equals the incident intensity Ii,

$$I_i = I_r + I_t \qquad (2.37)$$

And dividing throughout by Ii ,

$$\frac{Ir}{Ii} + \frac{It}{Ii} = R + T = 1 \qquad (2.38)$$

Where R is the reflection coefficient and T is the transmission coefficient. For plane waves at normal incidence on a plane boundary between two fluids (see Fig:2.8)

$$R = \frac{(\rho 1\ c1 - \rho 2\ c2)^2}{(\rho 1\ c1 + \rho 2 c2)^2} \qquad (2.39)$$

And

$$T = \frac{4\ \rho 1\ c1\ \rho 2\ c2}{(\rho 1\ c1 + \rho 2 c2)^2} \qquad (2.40)$$

Some interesting facts can be deduced from Eqs (2.39) and (2.40). Both the reflection and transmission coefficients are independent of the direction of the wave, since interchanging ρ1c1 and ρ2c2 dose not affect the value of R and T. For example, for sound waves traveling air to water or water to air, almost complete reflection occurs, independent of direction, and the reflection coefficients are the same and the transmission coefficients are the same the two different directions.

هناك معادلة نستخدمها في الحرارة تقول إن الكثافة الحادثة تساوي الكثافة النافذة يضاف إليها الكثافة المنعكسة. ومن جهة أخرى فإن معامل الإنعكاس مضافا إليه معامل النفاذية يساوي واحد.

ويستنتج ايضا من المعادلات السابقة أن معامل النفاذية أو معامل الإنعكاس كلا منهما لا يؤثر على إتجاه الموجة الصوتية

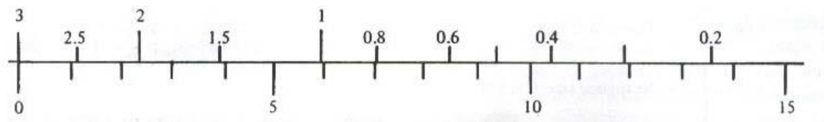


Fig: 2.8 Incident intensity Ii, reflected intensity Ir, and transmitted [4]

As discussed before, when the characteristic impedance ρc of a fluid medium changes, incident sound is both reflected and transmitted. It can be shown that if a plane sound wave is incident at an oblique angle on a plane boundary between two fluids then the wave transmitted into the changed medium changes direction. This effect is called refraction.

متى تغيرالموجة الصوتية إتجاهها؟

تغير المجة الصوتية إتجاهها بتغير الوسط الناقل أو تغير خصائصه. وكذلك تغير حرارة الجو وتغير سرعة الرياح.

Temperature changes and wind speed changes in the atmosphere are important causes of refraction. Wind speed normally increases with altitude, and Fig:2.9 shows the refraction effects to be expected for an idealized wind speed profile. Atmospheric temperature changes alter the speed of sound c, and temperature gradients can also produce sound shadow and focusing effect, as seen in Figs:2.10 and 2.11

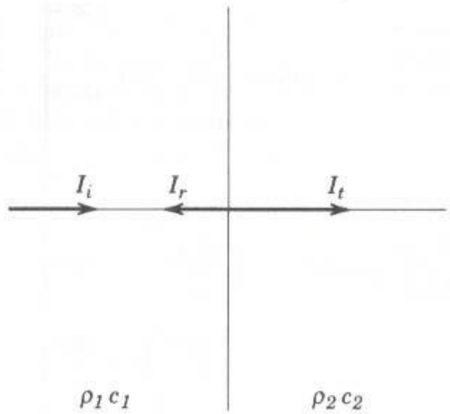


Fig:2.9 Refraction of sound in air with wind speed U(h) increasing with altitude h.[5]

Wind speed normally increases with altitude, and Fig:2.9 shows the refraction effects to be expected for an idealized wind speed profile. Atmospheric temperature changes alter the speed of sound c, and temperature gradients can also produce sound shadow and focusing effects, as seen in Figs:2.10 and 11.

When a sound wave meets an obstacle, some of the sound wave is deflected. The scattered wave is defined to be the difference between the resulting wave with the obstacle and the undisturbed wave without the presence of the obstacle. The scattered wave spreads out in all directions interfering with the undisturbed wave. If the obstacle is very small compared with the wavelength, no sharp-edged sound shadow is created behind the obstacle. If the obstacle is large compared with the wavelength, it is normal to say that the sound wave is reflected (in front) and diffracted (behind) the obstacle (rather than scattered)

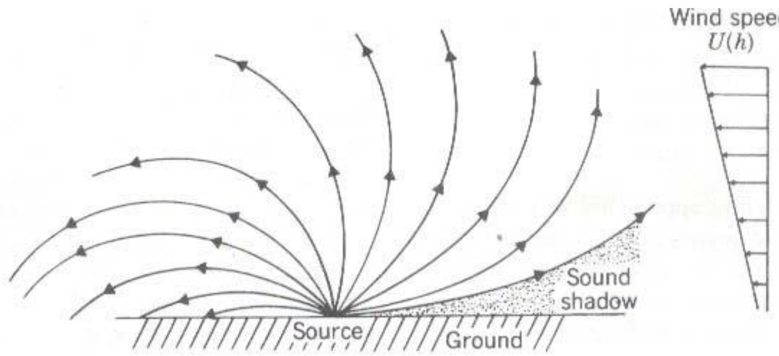


Fig:2.10 Refraction of sound in air with normal temperature lapse (temperature decreases with altitude) [5]

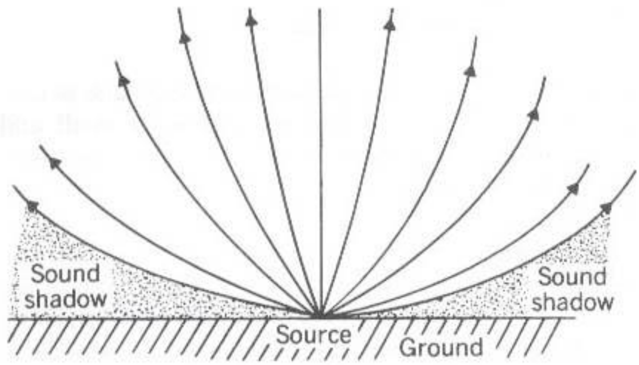


Fig:2.11 Refraction of sound in air with temperature inversion [5]

In this case a strong sound shadow is caused in which the wave pressure amplitude is very small. In the zone between the sound shadow and the region fully "illuminated" by the source the sound wave pressure amplitude oscillates. These oscillations are maximum near the shadow boundary and minimum well inside the shadow. These oscillations in amplitude are normally termed diffraction bands. One of the most common examples of diffraction caused by a body is the diffraction of sound over the sharp edge of a barrier or screen. For a plane homog-eneous sound wave it is found that a

strong shadow is caused by high- frequency waves where h/λ > 1 and a weak shadow where h/λ <1,

وتؤثر درجة الحرارة علـى شـكل إنتشـار الموجـة الصـوتية. (موضـح فـي الشـكل 2.11 ,2.10). وينشأ ظل الصوت أو بمعنى أدق بقايا الصـوت وأثـره في حالة التغير الحراري وفي حالـة وجـود حائـل لايمنـع الصـوت بمقـدار مائة في المائة. فإذا ما كان هذا الحائل أكبر مـن قيمـة الطـول المـوجي فإن الموجة الصوتية ترتد .

ويمكن الإنصياع فـي تقيـيم شـدة الصـوت المتبقـي (ظـل الصـوت) إلـى بعـدين فقـط وهمـا إرتفـاع الحائـل والطـول المـوجي. فـإن كـان حاصـل التقسـيم أكبر من واحد كان ظل الصوت قوي ، والعكس بالعكس.

Where h is the barrier height and λ is the wavelength. For intermediate case where h/λ ≈ 1a variety of interference and diffraction effects are caused by the barrier. Scattering is caused not only by obstacles placed in the wave field, but also by fluid regions where the properties of the medium such as its density or compressibility change their values from the rest of the medium. Scattering is also caused by turbulence and from rain or fog particles in the atmosphere and bubbles in water and by rough or absorbent areas on wall surfaces.

Ray acoustics

There are three main modeling approaches in acoustics, which may be termed **wave acoustics**, **ray acoustics**, and **energy acoustics**. So far in this chapter we have mostly used the wave acoustics approach in which the acoustic quantities are completely defined as functions

of space and time. This approach is practical in certain cases where the fluid is unbounded as long as the fluid is homogenous. However, if the fluid properties vary in space due to variations in temperature or due to wind gradients, then the wave approach becomes more difficult and other simplified approaches such as the ray acoustics approach described here This approach can also be extended to propagation in fluid-submerged elastic structures.

In the ray acoustics approach, rays are obtained that are solutions to the simplified eikonal eq.(2.41)

$$\left(\frac{ds}{dx}\right)^2 + \left(\frac{ds}{dy}\right)^2 + \left(\frac{ds}{dz}\right)^2 + \frac{1}{c^2} = 0 \qquad (2.41)$$

The ray solutions can provide good approximations to more exact acoustic solutions. In certain cases they also satisfy the wave equation. The eikonal s(x,y,z) represents a surface of constant phase (or wavefront) that propagates at the speed of sound c. It can be shown that Eq.41 is consistent with the wave equation only in the case when the frequency is very high. However, in practice, it is useful, provided the changes in the speed of sound c are small when measured over distance comparable with the wavelength. In the case where the fluid is homogeneous (constant speed C0 and density ρ throughout) S is constant and represents a plane surface given by S= (αx + βy + γz)/ c0, where α,β and γ are the direction cosines of a straight line (a ray) that is perpendicular to the wavefront (surface S). if the fluid can no longer be assumed to be homogeneous and the speed of sound c(x,y,z) varies with position, the approach becomes approximate only. In this case some parts of the wavefront move faster than others, and the rays bend and are no longer straight lines. In cases where the fluid is not stationary, the rays are no longer quite parallel to the normal to the wavefront. This ray approach is described in more detail in several books.

الأشعة الصوتية

تختلف الأشعة الصوتية عن الموجات الصوتية في أن واقع الأمر للموجة الصوتية أو كل نقطة من نقاط المنحنى السيني لها ثلاثة أبعاد X,Y,Z وقد تختلف معاملات تلك الإحداثيات لذا فإن معادلة 2.41 تترجم هذا الأمر ، يضاف إليها مقلوب مربع السرعة

The ray approach is also useful for the study of propagation in the atmosphere and is a method to obtain the results given in figs:2.9-2.11. It is observed in these figures that the rays always bend in a direction toward the region where the sound speed is less. The effects of wind gradients are somewhat different since in that case the refraction of the sound rays depends on the relative directions of the sound rays and the wind in each fluid region.

Energy acoustics

In enclosed spaces the wave acoustics approach is useful, particularly if the enclosed volume is small and simple in shape and the boundary conditions are well defined. In the case of rigid walls of simple geometry, the wave equation is used, and after the applicable boundary conditions are applied, the solutions for the natural (eigen) frequencies for the model (standing waves) are found. However, for large rooms with irregular shape and absorbing boundaries, the wave approach becomes impracticable and other approaches must be sought. The ray acoustics approach together with the multiple-image-source concept is useful in some room problems, particularly in auditorium design or in factory spaces where barriers are involved. However in many cases a statistical approach where the energy in sound field is considered is the most useful.

For a plane wave progressing in one direction in a duct of unit cross section, all of the sound energy in a column of fluid c meters in length must pass through the cross section in 1s. Since the intensity I is given by $P^2/\rho c$, then the total sound energy in the fluid column c meters long must also be equal to I. The energy per unit volume ε (joules per cubic meter)

Is thus

$$\varepsilon = \frac{I}{c} \qquad (2.42)$$

or

$$\varepsilon = \frac{p^2}{\rho c^2} \qquad (2.43)$$

The energy density ε may be derived by alternative means and is found to be the same as that given in Eq.(42)in most acoustic fields, except very close to sources of sound and in standing-wave fields. In a room with negligibly small absorption in the air or at the boundaries, the sound field created by a source producing broadband sound will become very reverberant (the sound wave will reach a point with equal probability from any direction). In addition, for such a case the sound energy may be said to be diffuse if the energy density is the same anywhere in the room. For these conditions the time-averaged intensity incident on the walls (or on an imaginary surface from on side) is.

$$I = \frac{1}{4}\varepsilon c \qquad (2.44)$$

or

$$I = \frac{p^2}{4\rho c} \qquad (2.45)$$

In any real room the walls will absorb some sound energy (and convert it into heat). The absorption coefficient α(f) of the wall material may be defined as the fraction of the incident sound intensity that is absorbed by the wall surface material :

$$\alpha = \frac{sound\ intensity\ absorbed}{sound\ intensity\ incident} \qquad (2.46)$$

The absorption coefficient is a function of frequency and can have a value between 0 and 1. The noise reduction coefficient (NRC) is found by averaging the absorption coefficient of the material at the frequencies 250, 500, 1000 and 2000 Hz (and rounding of the result to the nearest multiple of 0.05).

If we consider the sound field in a room with a uniform energy density ε created by a sound source that is suddenly stopped, then the sound pressure level in the room will decrease. We define a reverberation time in such a room as the time that the sound pressure level takes to drop by 60dB. We may show that the reverberation time TR is given as

$$T_R = \frac{0.163\ V}{S.\alpha} \qquad (2.47)$$

Where V is the room volume in cubic meters, S is the wall surface area in square meters, and α is the average absorption coefficient of the wall surfaces.

علـى جميـع مـن يتصـدر للعمليـات التصـميمية الخاصـة بالمسـارح ودور السـينما أن يعلم أن نقصان زمن الصدى عـن ثانيـة واحـدة أمـر ضـروري. والفراغ الجيد هو الذي ينعدم فيه زمن الصدى حتى يقتـرب مـن الصـفر. لكن مـا هـي العوامـل التـي تسـاعد علـى هـذا النقصـان.لكن مـا هـي العوامل التي تساعد على هذا النقصان؟

- زيادة المسطحات الماصة للصوت
- زيادة معامل الإمتصاص
- تبني عدم التوازي بين الأرضية والسقف أو بين جدارين متقابلين
- وجـود كاسـرات صـوتية المسـافة بينهـا تسـاوي عشـر متوسـط الطول الموجي.

By considering the sound energy radiated into a room by a broadband noise source of sound power W, we may sum together the mean squares of the sound pressure contributions caused by direct and reverbant fields after taking logarithms obtain the sound pressure level in the room:

$$L_p = L_W + 10\log\left(\frac{D\theta,\Phi}{4\pi r^2} + \frac{4}{R}\right) \qquad (2.48)$$

Where Dθ,Φ is the directivity factor of the source and R is the so-called room constant.

$$R = \frac{S\alpha}{1-\alpha} \qquad (2.49)$$

A plot of the sound pressure level against distance from the source is given for various room constants in Fig:2.12 it is seen that there are different regions. The near and far fields depend on the type of source and the free field and reverberant field. The free field is the

region where the direct term $D/4\pi r^2$ dominates and the reverberant field is the region where the reverberant term 4/ R in Eq.(48) dominates. The so called critical distance $r_c = (D_{\theta,\Phi} R/ 16\pi)^{1/2}$ occurs where the two terms are equal.

Sound radiation from idealized structures

There are interesting phenomena observed with free-bending waves. Unlike sound waves, these are dispersive and travel faster at higher frequency. **The bending- wave speed is $C_b = (\omega kc)^{1/2}$,** where k is the radius of gyration $h/(12)^{1/2}$.h is the thickness, and c is the longitudinal wave speed $\{E/[\rho(1-\sigma^2)]\}^{1/2}$, where E is youngs modulus of elasticity. When the bending-wave speed equals the speed of sound in air, the frequency (see Fig.13). The critical frequency is

$$f_c = \frac{c^2}{2\pi kc} \qquad (2.50)$$

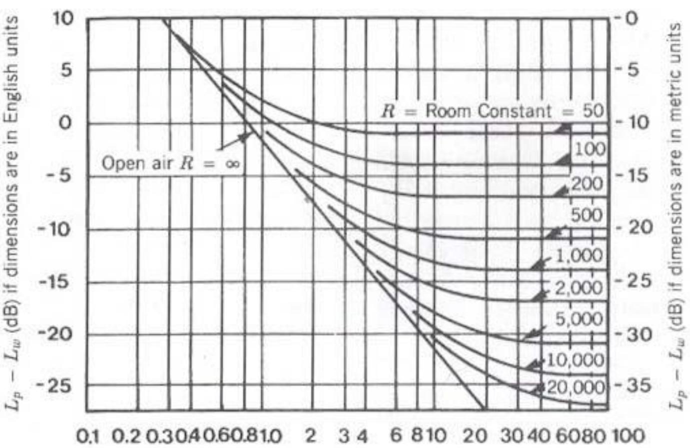


Fig:2.12 Sound pressure level in a room (relative to sound power level) as a function of distance ® [6]

Above this frequency fc the coincidence effect is observed because the bending wavelength λ is greater than the wavelength in air λ and trace wave matching always occurs for the sound waves in air at some angle of incidence (See fig:2.15). This has important consequences for the sound radiation from structures and also for the sound transmitted through the structures from one airspace to the other.

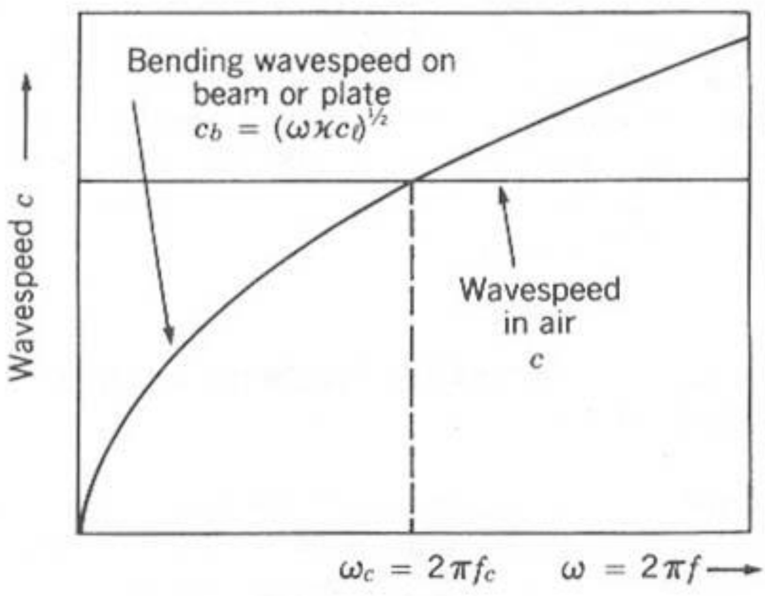


Fig:2.13 Variation of frequency of bending wave speed cb on a beam or panel and wave speed in air c [6]

For free-bending waves on infinite plates above the critical frequency the plate radiates efficiently, while below this frequency(theoretically) the plate cannot radiate any sound energy at all. For finite plates, reflection of the bending waves at the edges of the plates causes standing waves that allow radiation (although

inefficient) from the plate corner or edges even below the critical frequency.

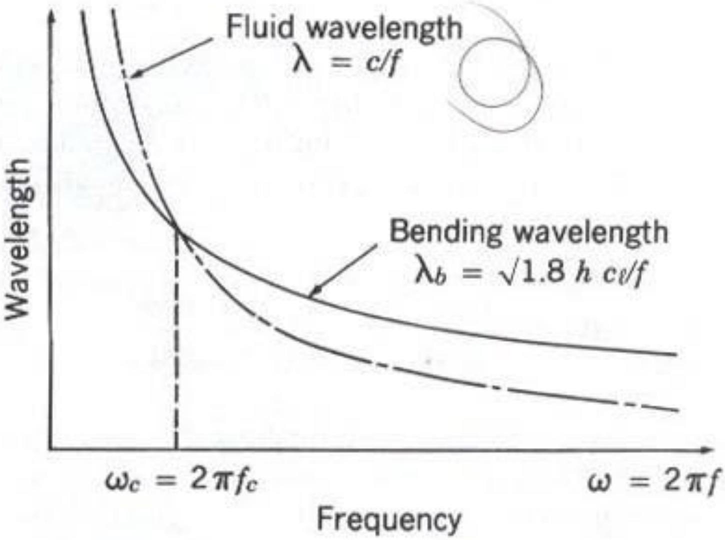


Fig:2.14 Variation with frequency of bending wavelength λb on a beam or panel and wavelength in air λ [6]

ماهي موجة الصوت المنثنية؟

هناك ظاهرة تحدث عند الترددات العالية وهي وجود موجات صوتية متبددة، سرعتها تكون أعلى من سرعة موجة الصوت العادية. بمعنى أن C_b أكبر من C . وبالمناسبة فإننا حين نختبر كفاءة مادة ما في عزلها للصوت نقول إذا كان التردد الرنيني أكبر من 1500 هيرتز تكون المادة صالحة للعزل وإذا كانت أقل تكون المادة غير صالحة للعزل. ولكن ماهي العلاقة بين التردد الرنيني والموجة المتبددة؟

العلاقة أن هذا التردد هو حاصل تقسيم مربع السرعة للموجة الطولية على الموجة المتبددة .والعلاقة علاقة طردية لوغاريتمية ،فالتناسب

بين السرعتين ، سرعة الموجة الصوتية في الهواء والموجة المتبددة على سطح الكمرة ، ليست خطية. انظر شكل 2.13

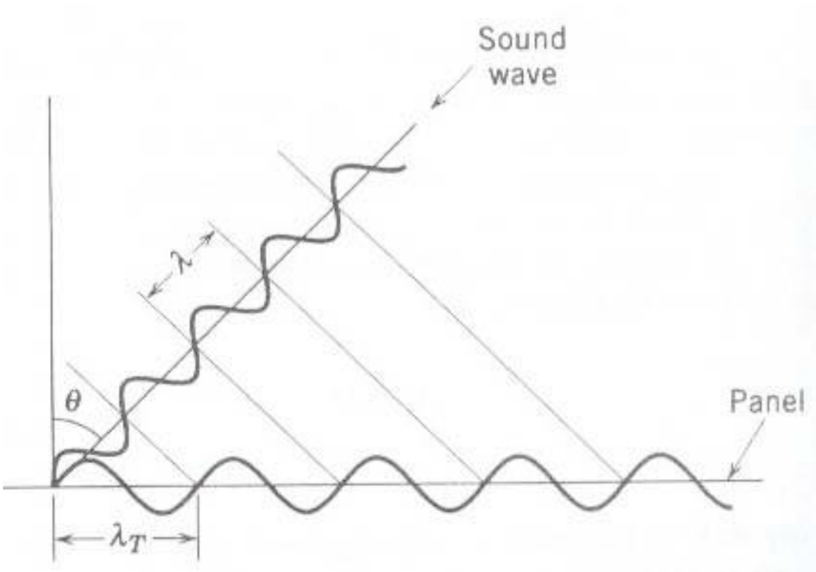


Fig: 2.15 Diagram showing trace wave matching between waves in air of wavelength λ and waves in panel of trace wavelength λt [7]

In the plate center, radiation from adjacent quarter-wave areas cancels. But radiation from the plate corners and edges, which are normally separated sufficiently in acoustic wavelengths, does not cancel. At very low frequency, sound is radiated mostly by corner modes, then up to the critical frequency mostly by edge modes. Above the critical frequency the radiation is caused by surface modes with which the whole plate radiates efficiently (see Fig.16).

وبعد أن علمنا العلاقة بين سرعة الموجة الطولية والموجة المتبددة تنعكس نفس العلاقة على الطول الموجي بين السرعتين (راجع 2.14) وعلينا أن نفرق بين سقوط الموجة على مركز سطح ما أو على حافة

هذا السطح ، فعادة ما يكون إنعكاس الموجة في الحافة أكبر ، أيا كان التردد، أي سواء في الترددات العالية أو الترددات المنخفضة

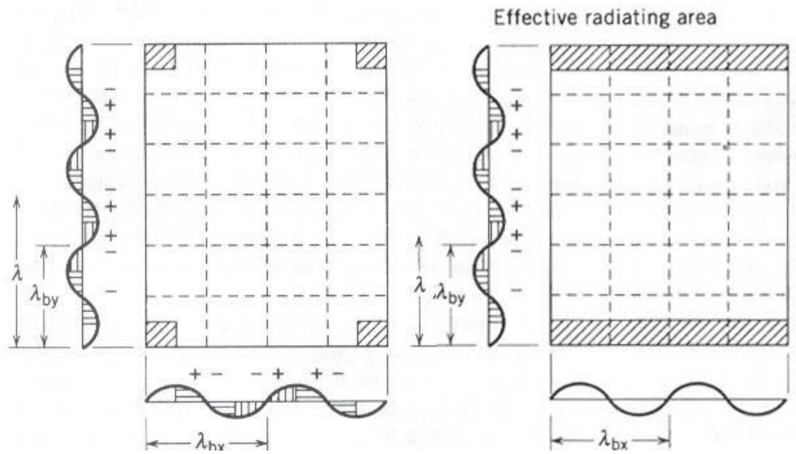


Fig: 2.16 Wavelength relations and effective radiating areas for corner, edge, and surface modes. The acoustic wavelength is λ., and λbx and λby are the bending wavelengths in the x- and y-directions, respectively[7]

<u>Boundary waves in acoustics</u>

Boundary waves in acoustic systems and elastic structures are observed on all scales – from the sub-millimeter (acoustic delay lines) to planetary size (Rayleigh modes in seismology). As their existence is essentially independent of that of the body or volume waves of acoustics or elasticity, they constitute a distinct and separate class of modes. The essential mechanism is one of storage at a boundary; the role of the boundary is intrinsic. In contrast, body or volume waves are in no way governed by the boundary, whose role is thus extrinsic. The energy density of boundary waves decreases exponentially or at a higher rate with increasing distance from the boundary. Reciprocally the excitation of such modes decreases exponentially

with the distance of the source from the boundary. Homogeneous plane waves incident on a plane boundary do not generate true surface modes. The theory and practice of acoustics and elasticity offer many examples of such modes: surface waves traveling along a rigid periodic structure of corrugations on a hard wall bounding a fluid half space, Stoneley waves along plane elastic-elastic interfaces, Scholte waves along plane fluid-elastic interfaces, boundary roughness modes Rayleigh waves on the surface of an elastic solid, and so on. Boundary waves exhibit the remarkable property of channeling spherically propagating energy produced by a point source into two-dimensional propagating configurations, that is, from r^2 geometric energy spreading loss to an r^{-1} law. That is why Rayleigh waves of shallow-focus earthquakes dominate seismograms at middle or long ranges- an effect also observed in the laboratory with boundary roughness modes in acoustics.

Standing waves

Standing-waves phenomena are observed in many situations in acoustics and the vibration of strings and elastic structures. Thus they are of interest with almost all musical instruments (both wind and stringed); in architectural spaces such as automobile and aircraft cabins; and in numerous cases of vibrating structures. From tuning forks, xylophone bars, bell and cymbals, to windows, wall panels and innumerable other engineering system including aircraft, vehicle, and ship structural members. With each standing wave is associated an eigen (or natural) frequency and mode a shape (or shape of vibration). Some of these systems can be idealized to simple one-, two-, or three-dimensional systems. For example with a simple wind instrument such as a whistle, Eq.(2.1) above together with the appropriate spatial boundary conditions can be used to predict the

predominant frequency of the sound produced. Similarly the vibration of a string on a violin can be predicted with an equation identical to Eq. (2.1) but with the variable P replaced by the lateral string displacement. With such a string, solutions can be obtained for the fundamental and higher natural frequencies (overtones) and the associated standing-wave mode shape (normally sine shapes).

الموجات الثابتة

تعرف الموجات الثابتة في جميع الآلات الموسيقية ، خاصة الوترية ، وفي كابينة السيارة والطائرة والعناصر الإنشائية المرنة. وفي جميع الأحوال سواء تحولت إلى إنتشار أحادي أو ثنائي أو ثلاثي الأبعاد فهي تأخذ الشكل السيني.

وحقيقة الموجات الثابتة أنها موجودة في الوتر أو الخيط المهتز ، والذي تخرج منه الموجة الصوتية على مرحلتين ، مرحلة ال standing wave ومرحلة التحول إلى sound wave in a sine shape والموجة الثابتة يمكن أن تكون عبارة عن موجتين بنفس سعة الموجة ولكن في الإتجاه العكسي.

In such a case for a string with fixed ends, the so-called overtones are just integer multiples (2,3,4,5,…) of the fundamental frequency. The standing wave with the whistle and string can be considered mathematically to be compressed of two waves of equal amplitude traveling in opposite directions.

A similar situation occurs for bending waves on bars, but because the equation of motion is different (dispersive), the higher natural frequencies are not related by simple integers. However, for the case of a beam with simply supported ends, the higher natural frequencies

are not given by 2^2, 3^2,4^2,5^2,..... or 4, 9, 16, 25, …. And the mode shapes are sine shapes again.

The standing waves on two-dimensional systems (such as bending vibrations of plates) may be considered mathematically to be composed of four opposite traveling waves. For simply supported rectangular plates the mode shapes are sine shapes in each direction. For three-dimensional systems such as the air volumes of rectangular rooms, the standing waves may be considered to be made up of eight traveling waves. For a hard walled room, the sound pressure has a cosine mode shape with a maximum pressure at the walls, and the particle velocity has a sine mode shape with zero normal particle velocity at the walls.

Waveguides

Waveguides can occur naturally where sound waves are channeled by reflections at boundaries and by refraction. The ocean can be considered to be an acoustic waveguide that is bounded above by the air-sea interface and below by the ocean bottom. Similar channeling effects are also some-times observed in the atmosphere. Waveguides are also encountered in musical instruments and engineering applications. In addition, waveguides comprised of pipes, tubes, and ducts are frequently used in engineering systems, for example, air conditioning ducts and ductwork in turbines and turbofan engines. The sound propagation in such wave guides is similar to the three-dimensional situation. Although rectangular ducts are used in air conditioning systems, circular ducts are also frequently used, and theory for these must be considered as well. In real waveguides, air flow is often present and complications due to mean fluid flow must be included in the theory.

For low-frequency excitation only plane waves can propagate along the waveguide (in which the sound pressure in uniform across the duct cross section). However as the frequency is increased, the so-called first cut-on frequency is reached above which there is standing wave across the duct cross section caused by the first higher mode of propagation.

For excitation just above this cut-on frequency, besides the plane-wave propagation in this higher order mode can be also exist. The higher mode propagation in each direction in a rectangular duct can be considered to be compressed of four traveling waves each with a vector (ray) almost perpendicular to the duct walls and with a phase speed along the duct that is almost infinite. As the frequency is increased, these vectors move increasingly toward the duct axis and the phase speed along the duct decreases until at very high frequency it is only just above the speed of sound c. However, for this mode the sound pressure distribution across the duct cross section remains unchanged. As the frequency increases above the first cut-on frequency, the cut-on frequency for the second higher order mode is reached, and so on. For rectangular ducts, the solution for the sound pressure distrution for the higher modes in the duct consists of cosine terms with a pressure maximum at the duct walls, while for circular ducts, the solution involves Bessel functions.

Acoustic lumped elements

When the wavelength of sound is large compared with the physical dimensions of the acoustic system under consideration, then lumped-element approach is useful. In this approach it is assumed that the fluid mass, stiffness, and dissipation distributions can be lumped together to act at a point, significantly the analysis of the

problem. The most common example of this approach is its use with the well-known Helmholtz resonator in which the mass of air in the neck of the resonator vibrates at its natural frequency against the stiffness of its volume. A similar approach can be used in the design of the loudspeaker enclosure and the concentric resonators in automobile mufflers in which the mass of the gas in the resonator louvers (orifices) vibrates against the stiffness of the resonator (which may not necessarily be regarded completely as a lumped element). Dissipation in the resonator louvers may also be taken into account.

Wave Propagation

In this chapter we give more detailed discussion of some basic concepts referred to chapter 1. of particular concern here is the mathematical embodiment of those concepts that underlay the wave description of acoustics.

This mathematical theory began with Messene, Galileo, and Newton and developed into its more familiar from during the time of Euler and Lagrange. Prominent contributors during the nineteenth century include Poisson, Laplace, Cauchy, Green, Stokes, Helmholtz, Kirchhoff and Rayleigh.

الصوت.. وفيزياء الصوت ليست إلا متغير يخضع بشكل غير مباشر لمبادئ الديناميكا الحرارية. لقد كانت بداية التاريخ العلمي عند نيوتن وجاليلو واستمر بعدها على يد جاوس وكيرشهوف ولابلاس واينشتاين . والحقيقة أننا نستفيد من خواص المواد وتحولها من صورة إلى أخرى ، وأهم من ذلك طاقتها الكامنة. وما الصوت إلا موجة وبالتالي طاقة شأنها في ذلك شأن الموجة الكهرومغناطيسية القادمة من الشمس.

The basic mathematical principles underlying sound propagation, whether through fluids or solids, include the principles of continuum mechanics, which include a law accounting for the conservation of mass, a law accounting for the changes in energy brought about by work and the transfer of heat. The subject also draws upon thermodynamics, upon symmetry considerations, and upon various known properties of the substances through which sound can propagate. The theory is inherently approximate, and there are many different versions, which differ slightly or greatly from each other in just what idealizations are made at the outset. The concern here is primarily with the simpler mathematical idealizations that have proven useful in acoustics, and in particular with those associated

with linear acoustics, but the discussion begins with the more nearly exact nonlinear equations and explains how the linear acoustics follow from them.

Conservation of Mass

The matter through which an acoustic wave travels is characterized by its density ρ, which represents the local spatial average of the mass per unit volume in a macroscopically small volume.

This density varies in general with position, generically denoted here by the vector x, and with time t. The material velocity v with which the matter moves is defined so that ρv is the mass flux vector within the fluid. The significance of the latter is that, were one to conceivence of a hypothetical stationary surface within the material with a local unit-normal vector n(x) pointing from one side to the other, then ρv.n gives the mass crossing this surface per unit time and per unit area of the surface. Here, also, a local average is understood; the material velocity v(x,t) associated with a point and time may be considerably different from the instantaneous velocity of a molecule in the vicinity of that point.

الضغط في السوائل والغازات يعادل ويساوي الإجهاد في المواد الصلبة، والضغط ليس إلا الكثافة على المسطح، كثافة الغاز أو السائل على المسطح. هذا الغاز أو السائل هو الوسط الذي تنتقل فيه الموجة الصوتية.

من جهة أخرى إذا ضربنا الكثافة في السرعة فإننا نحصل على الوزن. أو أن تكامل العاملين بمتعلق المساحة S والزمن T هو وزن المتجة.

The concept of a mass flux vector leads to the prediction that the net rate of flow of mass out through any hypothetical closed surface s within the material is the area integral of ρv.n (the volume v enclosed by this surface is sometimes referred to as a control volume. The conservation of mass requires that this area integral be the same as the rate at which the volume integral of ρ decreases with time so that

$$\int_s \rho v . n \, ds = - \frac{d}{dt} \int_v \rho dv \qquad (2.51)$$

The partial differential equation expressing the conservation of mass results, after application of Gauss's theorem to the surface integral with the recognition that the volume of integration is arbitrary, so that

$$\frac{dp}{dt} + v\,(\rho v) = 0 \qquad (2.52)$$

Where the partial derivative with respect to time implies that the position x is held fixed in the differentiation. An alternate form of the above partial differential equation is

$$\frac{d\rho}{dt} + \rho \,.\bar{V}\, v = 0 \qquad (2.53)$$

The Euler Equation

The local rate of material acceleration is identified as the total time derivative dv/dt of the material velocity. Consequently, the generalization of Newton's second law to a continuum requires that ρ dv/dt equal the force per unit mass exerted on the material.

The force on any small identifiable aggregate of material consists of a sum of a body force Fb and a surface force Fs. Possible body forces are gravitational or electromagnetic; the former is customarily expressed as

$$F_b = \int_v \rho \; g \; dv \qquad (2.54)$$

Where g is the acceleration associated with gravity. The surface force is expressible as

$$Fs = \int_s \sum_{i,j=1}^{3} \rho ij \; . ei . nj \; dS \qquad (2.55)$$

Where the ρij are Cartesian components of the stress tensor. Here the quantities ei are the unit vectors appropriate for the corresponding Cartesian components of the vector pointing outward to the enclosing surface S.

ويسمي الفيزيائيون على إختلاف مشاربهم إحداثيات الكتلة المتحركة والتي يختلف وزنها حتما بإختلاف السرعة وتغير الزمن بالإحداثيات الديكارتية.

The stress tensors Cartesian component ρij can be regarded as the ith component of the surface force per unit area on a segment of the surface of a small element of the continuum, when the unit outward

normal to the surface is in the jth direction. This surface force is caused by interactions with the neighboring particles just outside the surface or by transfer or momentum due to diffusion of molecules across the surface. The condition that the net force per unit volume be finite in the limit of macroscopically infinitesimal volumes requires that the stress components ρij be independent of the shape and orientation of the hypothetical surface s, so the stress is a tensor field associated with the material that depends in general on position and time. Consideration of the net torque that such surface forces would exert on a small element and the requirement that the angular acceleration of the element be finite in the limit of very small element dimension lead to conclusion ρij=ρji, so the stress tensor is symmetric: with the above identifications for the surface and body forces and a subsequent application of Gauss theorem to transform the surface integral to a volume integral, the application of Newtons second law yields Cauchy equation of motion, which is written in Cartesian coordinates as:

$$\rho \frac{dv}{dt} = \sum_{ij} ei \ \frac{d\rho ij}{dxj} + g.\ \rho \qquad (2.56)$$

Here the Eulerian description is used, with each field variable regarded as a function of actual spatial position coordinates and time.

For disturbances in fluids, such as air and water, and with the neglect of viscous shear forces, the surface force associated with stress can only be normal to the surface. Since such would have to hold for all possible orientations of the surface, the stress tensor has to have the form

$$\rho ij = p\delta ij \qquad (2.57)$$

where p is identified as the pressure and where δij, equal to unity when the indices are equal and zero otherwise, is the kronecker delta. The pressure p is understood to be such that –pn is the force per unit area exerted on the material on the interior side a surface when the outward unit-normal vector is n. the insertion of the above idealization for the stress into cauchys equation of motion yields Eulers equation of motion for a fluid

$$\rho \frac{D\vec{v}}{Dt} = - \nabla p + g\rho \qquad (2.58)$$

Although viscosity is often important adjacent to solid boundaries and can lead to the attenuation of sound excellent first approximation to use the Euler equation as the embodiment of Newtons second for a fluid in the analysis of acoustic processes. Also, gravity has a minor influence on sound and can often be ignored in analytical studies.

The Euler equation can be alternately written, via a vector identity, as

$$\frac{dv}{dt} - v(\bar{\nabla}.v) + \bar{\nabla}\left(\frac{v^2}{2}\right) = -\frac{1}{\rho}\bar{\nabla}p + g \qquad (2.59)$$

This equation governs the dynamics of the vortices ($\bar{\nabla}$.v) and provides a means for assessing the common idealization that the vortices are identically zero.

Thermodynamic Principles in Acoustics

The quasi-static theory of thermodynamics presumes that the local state of the material at any given time can be completely described in terms of a relatively small number of variables. One such state variable is the density ρ; another is the internal energy u per unit mass (specific internal energy) .

وإذا كان الحديث عن الكثافة وعن الطاقة في وجود متغيرات كالضغط والحرارة والطاقة الداخلية الكامنة فإننا نتحدث عن الشكل العشوائي للطاقة Entropy ، والتي تتعلق بجميع تلك المتغيرات المذكورة يضاف إليها الكثافة والوزن.

For a fluid in equilibrium, these two variables are sufficient to specify the state. Other quantities such as the pressure p and the temperature T can be determined from equations of state that give these in terms of ρ and u. An important additional state variable, although one not easily measurable, is the equilibrium entropy s(x,t) per unit mass (specific entropy), the changes of which can be related to heat transfer, but which also can be regarded as a function of u and ρ. The physical interpretation of s requires that there be no heat transfer when s is constant, and conservation of energy requires that work done by pressure forces on a surface of a fluid element during a quasi-static process result in a corresponding increase in internal energy ρVdu. For a sufficiently small element, this work is p time the net decrease, -dv, in volume. Since mass is conserved, ρdv = -vdρ and the work is consequently pρ^{-1} vdρ. This yields ρvdu= pρ^{-1}dρ when ds=0, and leads to the prediction that, to first order in differentials, ds must be proportional to du-pρ^{-2}dρ , where the proportionality

factor must be a function of u and ρ > there is an inherent arbitrariness in the actual form of the function s(u,ρ) , but the theory of thermodynamics requires that it can be chosen so that the reciprocal of the proportionality factor has all the properties commonly associated with absolute temperature. Such a choice yields

$$T_{ds} = \quad du - p\,\rho^{-2}\,d\rho \qquad (2.60)$$

With the identification of T_{ds} as the increment of heat energy that has been transferred per unit mass during a quasi-static process. For an ideal gas with temperature-independent specific heats, which is a common idealization for air, the function s(u,ρ) is given by

$$S = \frac{Ro}{M}\ \ln(u^{1/(\Upsilon-1)}\rho^{-1}) + So \qquad (2.61)$$

Here the constant So is independent of u and ρ^{-1}, while M is the average molecular weight (average molecular mass in atomic mass units) , and Ro is the universal gas constant (equal to Boltzmann constant divided by the mass in an atomic mass unit), equal to 8314J/kgk . the quantity ϒ is the specific heat ratio, equal to approximately 7/5 for diatomic gases, 5/3 for monatomic gases, and 9/7 for polyatomic gases whose molecules are not collinear.

خصوصية الانتروبي

أعتذر للقارئ الكريم استعمالي في هذه الفقرة لفظا يونانيا مكتوب بالأحرف العربية، لكن لفظ الإندماج العشوائي ، والذي هو الترجمة العربية ، لن يكون معبرا عما نقول. الانتروبي معناها التحول وهو مفهوم هام في التحريك الحراري خاصة في القانون الثاني للديناميكا الحرارية.

وقبل أن أشير إلى الربط بينه وبين الموجات الصوتية لابد من الوقوف على بعض النقاط.

إن نظام الانتروبي يتعامل مع العمليات الفيزيائية للأنظمة الكبيرة المكونة من جزيئات بالغة الأعداد ويبحث سلوكها كعملية تتم بصورة تلقائية أو غير تلقائية.

إن أي نظام مغلق يميل إلى التغير أو التحول تلقائيا بزيادة انتروبيه حتى يصل إلى حالة توزيع متساوي في جميع اجزائه ، مثل تساوي درجة الحرارة وتساوي الضغط وتساوي الكثافة وغير تلك الصفات.

ويما أن الموجات الصوتية تعمتد على الضغط والكثافة ودرجة الحرارة أيضا ، إذا فشك انتشارها وطموحها في الوصول إلى حالة التساوى له علاقة بالإنتوبيه.

وقد يحتاج النظام المعزول في الوصول إلى هذا التساوي إلى بعض الوقت ، مثال ذلك إلقاء قطرة من الحبر الأزرق في كوب ماء ، نلاحظ أن قطرة الحبر تذوب وتنتشر رويدا رويدا في الماء حتى يصبح كل جزء في الماء متجانسا بما فه من حبر وماء ، فنقول إن انتروبية النظام تزايدت. أي إن مجموع انتروبية نقطة الحبر النقية + انتروبية الماء النقية تكون أقل من انتروبية النظام "حبر ذائب في ماء "

وقد أصبح للإنتروبية كأحد الصفات الطبيعية أهمية بالغة من خلال علاقة الانتربية بتحول الطاقة الحرارية إلى شغل ميكانيكي ، فنجدها تلعب دورا هاما في تحديد كفاءة الآلات ، كمحرك الإحتراق الداخلي ومحرك الديزل وغيرها.

وهنا لابد من الإشارة إلى أن الكمون الكيميائي ضمن أي نظام فيزيائي أو كيميائي يميل تلقائيا إلى خفض الطاقة الداخلية للنظام إلى أقل ما يمكن، لكي يصل النظام لحالة من التوازن. الانتروبي ضمن هذا المفهوم هو مقدار تقدم عملية التحول والتوازن هذه.

الانتروبي في الديناميكا الحرارية.

في الديناميكا الحرارية نقوم بوصف التبادل الحراري (تبادل طاقة) بين النظام والوسط المحيط. وتوجد امكانيتان للتفاعل بين النظام والوسط

المحيط ، فإما أن يتم عملية تبادل حراري بينهما أو شغل . ويتعامل القانون الأول للديناميكا الحرارية مع مصطلح الطاقة التي تكون محفوظة ضمن نظام فيزيائي مغلق. في نفس الوقت تعرف الانتروبي على أنها تغير وتحول إلى حالة أكثر فوضوية وهرجلة (مثال انتشار نقطة الحبر في الماء)على المستوى الجزئي في نظام، فالتغيرات التلقائية تميل دوما لكسب مزيد من الحرية لحركة الجزيئات أو الذرات. فإذا تخيلنا قارورتين تحتوي كل منهما على غاز غير الآخر وفتحنا بينهما فتحة، نجد أن الغازين يبدأن الانتشار العشوائي في القارورتين. وبعد فترة من الزمن يصل النظام إلى حالة اتزان وتساوي في توزيع الجزيئات، أي اذا اخذنا أي سنتيمتر مكعب من مخلوط الغاز من أي مكان في القارورتين فسوف نجد عددا متساويا من نوعي جزيئات الغاز المخلوطين.

ونتصور الآن أننا نريد فصل الغازين المختلطين في مثالنا السابق الذي هو مثال لعملية غير عكسية ، سوف نجد أنه عمل مضني ، فلا يمكن عكس مسار زيادة انتروبيا النظام إلا بأداء شغل .

$$\mathbf{dS} = \frac{dQ\ rev}{T}$$

حيث dS هي الانتروبي وتمثل الطاقة في النظام الفيزيائي ووحدتها جول / كلفن.

و Q تمثل التغير في مقدار حرارة النظام.

و T هي درجة الحرارة

For air, ϒ is 1.4 and M is 29.0 (The expression given here for entropy neglects the contribution of internal vibrations of diatomic molecules, which cause the specific heats and ϒ to depend slightly on temperature.

In the explanation of the absorption of sound in air, a non-equilibrium entropy is used that depends on the fractions of O2 and N2 molecules that are in their first excited vibrational states in addition to the quantities u and ρ^{-1} . With the aid of differential relation, the corresponding expression for temperature and pressure appear as

$$T = (\Upsilon - 1)\frac{M}{Ro}\,u \qquad (2.62)$$

$$P = (\Upsilon - 1)\rho\, u \qquad (2.63)$$

And from these two relations emerges the ideal-gas equation

$$P = \rho.\frac{Ro}{M}.T \qquad (2.64)$$

For other substances, a knowledge of the first and second derivatives of s , evaluated at some representative thermodynamic state, is sufficient for most linear acoustic applications.

An immediate consequence of the above line of reasoning is that for a fluid in equilibrium, any two state variables are sufficient to determine the others. The pressure, for example, can expressed as

$$P = p\,(\rho, s) \qquad (2.65)$$

Where the actual dependence on ρ and s is intrinsic to the material. For an ideal gas, for example, this function has the form

$$P = k(s).\rho^{\Upsilon} \qquad (2.66)$$

Where

$$K(s) = (\Upsilon-1)P0\ \rho_0^{-\Upsilon} \quad \exp\left((\Upsilon-1)\frac{M}{Ro}(s-s0)\right) \qquad (2.67)$$

Is a function of specific entropy only

The formulation of acoustics that is most frequently used, and with which the present chapter is concerned, assumes that the processes of sound propagation are quasi-static in the thermodynamic sense. This implies that the p given by equation of state (2.65) is the same as that which appears in the Euler equation (2.58). Also equation (2.61) must hold for any given possibly moving element of fluid, so that the differential relation yields the partial differential equation

$$T\frac{Ds}{Dt} = \frac{Du}{Dt} - P\ .\ \rho^{-2}\frac{D\rho}{Dt} \qquad (2.68)$$

With the neglect of heat transfer mechanisms within the fluid, the conservation of energy requires that

$$\rho\frac{D}{Dt}\left\{\frac{v^2}{2} + u\right\} = -\bar{V}\ .\ \{pv\} + \rho g.v \qquad (2.69)$$

where the quantity within braces on the left is the energy (kinetic plus potential) per unit mass and the terms on the right correspond to the work done per unit volume by pressure and gravitational forces , respectively. The Euler equation (2.58) yields the derivable relation

$$\rho\frac{D}{Dt}\left\{\frac{v^2}{2}\right\} = -v.\ \bar{V}p + g\rho.v \qquad (2.70)$$

and the subtraction of this from eq.(2.69)

$$\rho \frac{Du}{Dt} = - p. \overline{\mathrm{V}}.v = p\rho^{-1} \frac{D\rho}{Dt} \qquad (2.71)$$

The latter equality results from the version (3) of the conservation of mass equation. Therefore, the conservation-of–energy relation(2.69) requires that the right side of eq.(68) be identically zero, so

$$\frac{Ds}{Dt} = 0 \qquad (2.72)$$

This implies that the entropy per unit mass of any given fluid particle remains constant. A disturbance satisfying this relation is said to be isentropic. The total time derivative of eq (2.65) with the replacement of Ds/Dt by 0 yields the equation

$$\frac{Dp}{Dt} = c^2 \frac{D\rho}{Dt} \qquad (2.73)$$

Where c^2abbreviates the quantity

$$c^2 = \frac{dp}{d\rho} \qquad (2.74)$$

Here the subscript s implies that the differentiation is carried out at constant entropy. Thermodynamic considerations based on the requirement that the total entropy increases during an irreversible process requires in turn that this derivative be positive, its square root c is identified as the speed of sound, for reasons discussed in a subsequent portion of this chapter.

The relation between the total time derivatives of pressure and density that appears in eq. (2.73) can be combined with the conservation-of-mass relation to yield

$$\frac{Dp}{Dt} + \rho c^2 \bar{\nabla}.v = 0 \qquad (2.75)$$

Which is often of greater convenience in acoustics than the mass conservation relation because the interest is typically in pressure fluctuations rather than in density fluctuations? For an ideal gas, the differentiation of the expression p(ρ,s) in eq (65)yields

$$C^2 = \Upsilon\, k\, (s)\, \rho^{\Upsilon-1} \qquad (2.76)$$

Which can equivalently be written

$$C^2 = \frac{\Upsilon p}{\rho} = \Upsilon \frac{Ro}{M} T \qquad (2.77)$$

For air, R_0/M is 287 J/kgk and ϒ is 1.4 so temperature of 293.16 k (20°) and pressure of 10^5pa yield a sound speed of 343m/s and a density ρ of 1.19 kg/m^3. Acoustic properties of materials are discussed in depth, but some simplified formulas for water are given here as an illustration of the thermodynamic dependences of such properties. For pure water, the sound speed is approximately given in metreskilograms- second (MKS) unit by

$$C = 1447+4.0\ \Delta T+1.6x10^{-6} \qquad (2.78)$$

Here c is in meters per second ΔT is the temperature relative to 283.16k (10), and p is the absolute pressure in Pascals. The pressure

and temperature dependence of the density is approximately given by

$$\rho = 999.7 + 0.048 \times 10^{-3}\, p - 0.088\Delta T - 0.007\Delta T \qquad (2.79)$$

The values for sea water are somewhat different because of the presence of dissolved salts. An approximate expression for the speed of sound in sea water is given by

$$C = 1490 + 3.6\, \Delta T + 1.6 \times 10^{-6}\, p + 1.3\Delta S \qquad (2.80)$$

Here ΔS is the deviation of the salinity in parts per thousand from a nominal value of 35

Room Acoustics

The hearing process

Since acoustics has meaning only through our interpretation of sound, it is helpful to understand the human hearing mechanism to fully appreciate the science of acoustics. Fig:2.17 shows a cross section of the human hearing mechanism, divided into its most commonly labeled sections of the outer, middle, and inner ears.

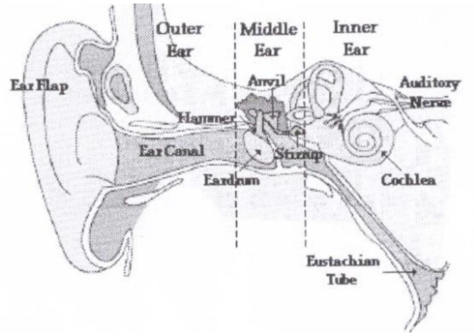


Fig:2.17 outer, middle, and inner ears [16]

The function of the outer ear is to funnel sound waves into the rest of the hearing mechanism. Without a pinna on the outside of our heads, we would not be able to hear most of the sounds around us. The picture of the mean with an inverted horn held up to his ear accentuates this point. The ear canal funnels the sound to the eardrum, also known as the tympanic membrane in medical circles.

The eardrum is the first location in the hearing mechanism where sound energy becomes converted into another form or energy in its trip to the interpretation center of the brain. The eardrum also represents the limit of the outer ear.

Changes in the direction of sound travel

Sound waves change their direction of travel through four categories of phenomena: Reflection, refraction, diffraction, and diffusion.

These phenomena can occur when changes occur in the sound waves medium of travel. These physical principles are the same as those that occur in the optical, world with light. The principal difference between light and sound is the frequency range. Our visible frequency range is 1.6 to 2.8 billion Hz, while our audible frequency range is 20 to 20000 Hz.

<u>ابداع الخالق</u>

إن ابداع الخالق ليس له حدود، ولا نريد أن نتطرق بالسرد إلى العديد من النعم. فالمقام ليس مقام إستدلال ولكن الإشارة التي ينبغي أن نتطرق إليها هي أن حدود الحاسة السمعية لنا من 20 إلى 20000 هيرتز. بمعنى أننا كمخلوقات لا نستطيع أن نسمع فوق ذلك ولا نستطيع كذلك أن نسمع أقل من ذلك.

وبما أن حاسة السمع محدودة فهل حاسة الإبصار محدودة كذلك؟

الإجابة نعم .. دون أدنى شك وليس هذا خاص بالنظر فقط بل بجميع الحواس البشرية ، يضاف إلى ذلك العقل الذي يتفكر ويسترشد لكي يهتدي ، هو محدود كذلك. ولكن ما الحكمة في هذه المحدودية ، لماذا السمع محدود؟ لأنه لو سمع الإنسان كل شيء حتى دبيب النمل لأصابه الجنون. ونفس الشيء في جميع الحواس الأخرى.

<u>Reflection</u>

When a sound wave encounters a sharp discontinuity in the density of a medium, some of its energy is reflected, as is illustrated in figure 2.18 reflected sound energy follows the laws of optics, with the simplest visualization analogy being a mirror. Just as light bounces off a mirror at the same angle as its angle of incidence, sound waves

have equal angles of incidence and reflection. Reflective surfaces are typically smooth and hard.

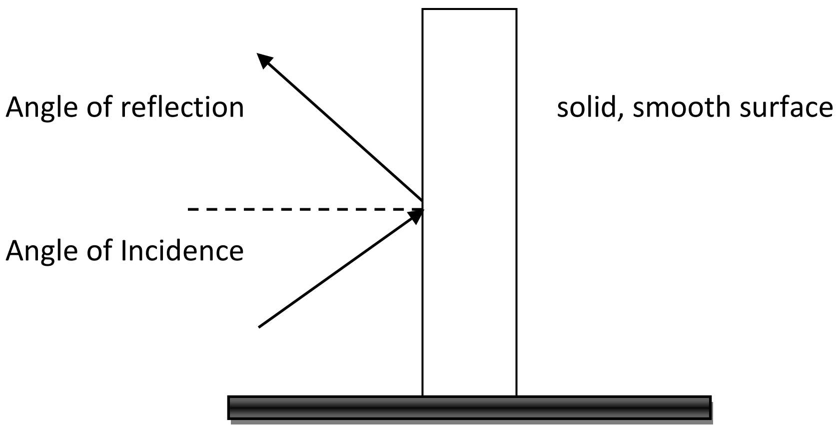


Fig: 2.18 Reflection and angle

<u>Refraction</u>

Just as light bends as it travels through a prism, the direction of sound is altered when sound waves encounter changes in medium conditions that are not extreme enough to cause reflection, but are enough to change the speed of sound. In addition to the speed of sound changing for different materials or media, the speed of sound changes with changes in temperature within the same medium. This variation in sound travel direction, caused by variations in the speed of sound, is known as refraction.

الموجة السمعية تمتص وتنعكس وتنفذ من خلال وسط ما للوصول إلى فراغ آخر. والموجة السمعية تتغير سرعتها بتغير الوسط أو بتغير درجة الحرارة في نفس الوسط . سبق معنا هذا الكلام .

Diffraction

The principle of diffraction limits the sound reduction effectiveness of any open-plan office partition or outdoor noise barrier. Sound waves bend around and over these types of walls, independent of their material, to imposing (as with light) what is known as a shadow zone after the line of sight is broken between a sound source and listener

Diffusion

When a sound wave reflects off a convex or uneven surface, its energy is spread evenly rather than being limited to a discrete reflection. This phenomenon, known as diffusion (see Figure 19)

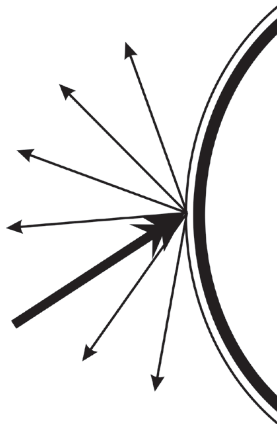


Fig: 2.19 diffusion of sound waves

Is the acoustic equivalent to the diffusion of the light from a frosted bulb rather than a clear bulb? Although discrete reflections in the form of the echoes are usually unwanted, it may not be desirable to eliminate that sound energy in a room. For example, diffusion can be useful in an auditorium or concert facility to spread sound evenly

throughout an audience and ensure that all audience members heir the same sound quality. This minimizes the potential for bad seats, at least from an acoustical standpoint.

The Decibel

One of the most misunderstood aspects of acoustics is the description of sound in terms of decibels (usually denoted as dB). To understand decibels, there are two basic points that must be established:

1- A decibel is defined in terms of logarithmic ratio. Logarithms are exponents of 10. Therefore, the logarithm of 10 is 1 (since 10 is 10 to the first power), the logarithm of 100 is 2 (since 100 is 10 to the second power), and the logarithm of 1000 is 3 (since 1000 is 10 to the third power).
2- The ratio in the decibels definition has a value that refers to a type of unit, such as power or pressure or intensity. This unit must be specified for a decibel value to have meaning.

Sound control

When we talk about controlling sound, it is often assumed that we are referring to the reduction of sound. However, there are cases where we want to preserve the sound energy but we would like to control its spatial spreading characteristics. The primary ways to reduce sound are through absorption and insulation. Using absorption on an auditoriums side walls may eliminate unwanted reflections but may also eliminate the possibility of some people

hearing sound coming from the stage. Therefore we must clarify how to control the sound.

Redirection and diffusion can have favorable acoustic results for even sound distribution in large rooms. On the other hand, discussions about noise control usually refer to the reduction of sound.

Fig:2.20 shows what happens to a sound wave that interacts with a room's surface. Part of it is absorbed appear in form of heat, part is redirected, and the rest is transmitted through the surface.

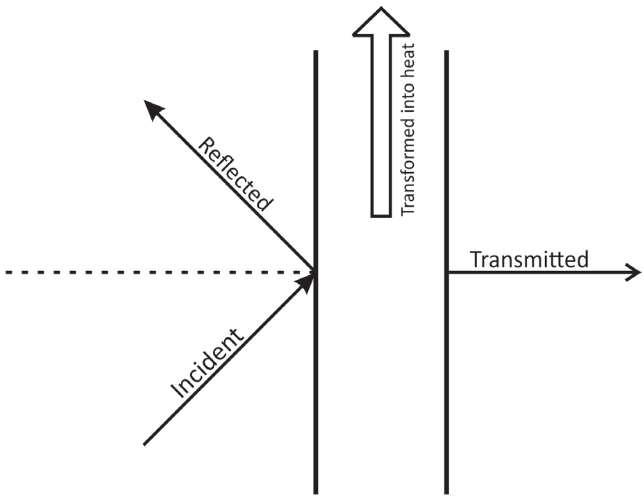


Fig:2.20 Absorption of sound waves

وبما أننا نؤمن بالقانون الأول للديناميكا الحرارية ونؤمن كذلك بقانون حفظ الطاقة فإن الجزء الممتص من الموجات لايختفي ولا يتلاشى ولكنه يتحول إلى صورة أخرى من صور الطاقة وهي الطاقة الحرارية.

ويسأل سائل فيقول هل معنى هذا أن الفراغ الذي يكثر فيه الحديث يكون أكثر سخونة من غيره، أقول نعم ولكن مقدار الحرارة الضئيل الناتج عن عملية تحول الموجات السمعية لايشعر الإنسان بها حسيا.

Absorption

Absorption converts sound energy into heat energy and is used to reduce sound levels within rooms. It is not effective in reducing sound between rooms. Each material with which a sound wave interacts absorbs some sound.

The most common measurement of that is the absorption coefficient. The absorption coefficient is a ratio of absorbed to incident sound energy if a material does not absorb any sound incident upon it, its absorption coefficient is 0

In practice, all materials absorb some sound, so this is a theoretical limit. If a material absorbs all sound incident upon it, its absorption coefficient is 1.

As with the lower limit for absorption coefficients, all materials reflect some sound, so this is also a theoretical limit. Therefore, the limits of absorption coefficient are 0 and 1.

وتحول الموجة السمعية إلى حرارة لا يحـدث إلا فـي عمليـة الإمتصـاص للموجة. أما في الإنعكاس أو الإنحـراف للموجـة الصـوتية فـلا يحـدث أي تحول إلى الحرارة.

أما بالنسـبة لمعاملات الإمتصاص بإختلاف المادة الممتصـة فهـي تتـدرج من صفر إلى واحد. وحاصل ضـرب هـذا المعامـل فـي المسـطح يوصـلنا إلى المسطح الفاعل .

Insulation

The description of the insulation of sound is similar in many ways to the description of the absorption of sound. As for absorption, there is a transmission coefficient that ranges from the ideal limits of 0 to 1.

The transmission coefficient is the unit less ratio of transmitted to incident sound energy.

Unlike the absorption coefficient, however, the limit of 1 is practically possible since a transmission coefficient of 1 implies that all of the sound energy is transmitted through a partition.

لي بحث قيم نشر في مجلة العلوم الأمريكية american science journal في المقارنة بين مسار الحرارة ومسار الصوت داخل الجدار. انتهى البحث إلى أن وزن الجدار يؤثر في انتقال الصوت بينما تساوى طبقات العزل مع الطبقة المفرغة هوائيا airgap layer في العزل الحراري.

وعليه فإن الذي يؤثر على عملية النقل في الجانبين قد يكون الوزن وقد يكون تصميم الطبقات وقد يكون أيضا طبيعة المادة من حيث معاملات النقل.

This would be the case for an open window or door, where the sound energy has no obstruction to its path. The other extreme of Zero (implying no sound transmission), however, is not a practical value since some sound will always travel through a partition.

Unlike absorption, the principal descriptor for the sound insulation is a decibel level based on the transmission loss (TL) is based on the logarithm of the mathematical reciprocal of (1 divided by) the transmission coefficient. The transmission loss can be loosely defined as amount of sound reduced by a partition between a sound source and listener.

The complete sound reductions of a partition between two rooms also take into account the partitions installation and the absorptive characteristics of the rooms. However, TL is the quantity that is

typically reported in manufacturer's literature since it is measured in the laboratory independent of the installation.

Since the logarithm of 1 is 0, the condition in which the transmission coefficient is 1 translates to TL of 0. This concurs with the notion that an open air space in a wall allows the free passage of sound. The practical upper limit of TL is roughly 70dB. Multilayered partitions comprise layers of different material, it is reduced. Therefore, this method can be used to reduce costs and space restrictions while providing adequate sound reduction. A sharp change in density of materials is most effective in this manner. Air spaces between sections and materials are effective by setting up such environments and by breaking any rigid connections between sides of a partition. A rigid connection can provide a vibration channel for sound to pass through with little reduction. For example, the noise reduction effectiveness of a studded wall filled with fiber insulation between studs can be short-circulated because sound will travel through the studs to the other side of the wall, as shown in fig:2.21

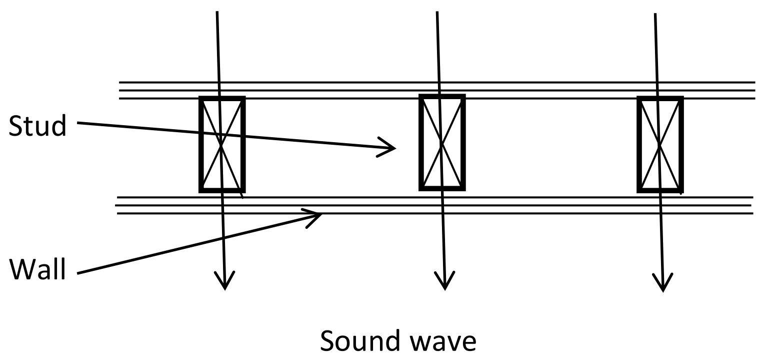


Fig:2.21 Waves Path

Staggered studs for the same wall can provide significantly higher TL while sacrificing minimal space.(Fig:2.22)

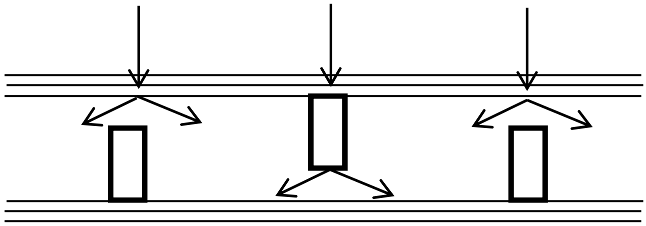


Fig:2.22 Waves Path

Noise Reductions Methods

Noise by definition is unwanted sound. We therefore want to eliminate, rather than redirect, noise when we talk about controlling it. Noise can be controlled at its source, in the path between the source and the listener.

If the noise can be controlled at its source, it is unnecessary to consider the path or listener locations. Likewise, if the noise can be controlled in the path between the source and listener, it is unnecessary to consider the listeners location for the noise control measures.

The options for noise control at the source are generally self-explanatory. Although they are the preferred noise control options, they are often impractical logistically or economically. Most often, noise control options are limited to the path between the source and the listener and at the listener driven many misconceptions about these options, it is useful to discuss some of them further.

Room Acoustics

The general considerations

- Size-minimize the room volume where low reverberation times are necessary (speech auditoriums) and choose the proper larger room volume for cases where medium or high reverberation times are required (halls for music)
- Absorption-add absorptive materials to reduce reverberation and add reflective or diffusive materials to add reverberation.
- Low-frequency absorption-use Helmholtz resonators, diaphragmatic absorbers, or plenum absorbers for large rooms having reflective surfaces and in need of speech intelligibility.
- Speech intelligibility-for large rooms, use a distributed sound system with appropriate delays between loudspeakers and focused low-level loudspeaker system for reverberant rooms. low-level loudspeaker systems for reverberant rooms.
- Sound systems are often used in rooms where contemporary music is being played. While we would like the natural acoustics of the room to provide the necessary sound quality, these sound systems are often set at such high levels that they negate the effect of any architectural designs. Artificial reverberation can also be set in these sound systems. It is therefore advisable to have as much absorption as possible in rooms where contemporary amplified music will be played.

Mathematical Expressions

Relationship between frequency and wave wavelength

$$C = f \cdot \lambda \qquad (2.81)$$

Where C= speed of sound in (ft/sec or m/sec)

f = frequency (inHz)

λ = wavelength (in ft or m)

Decibel definitions

Basic dB = $10 \times \log \frac{L}{L\ ref}$

Where L = sound power

Lref = reference sound power (1×10^{-12} watts)

Sound pressure level: Spl = $20\log\left(\frac{p}{Pref}\right)$

Where P = acoustic pressure

Pref = reference acoustic pressure (2×10^{-5} N/m^2)

at the threshold of hearing

Reverberation time $T = 0.162\ \frac{V}{A}$

Where V = room volume (m^3)

T = total room absorption = $\alpha_1 S_1 + \alpha_2 S_2 + \ldots\ldots$

Where α = absorption coefficients for different room material

S = surface area

Transmission loss

$$T_L = 10 \log \left(\frac{1}{\tau} \right)$$

Where TL = transmission loss

τ = transmission coefficient

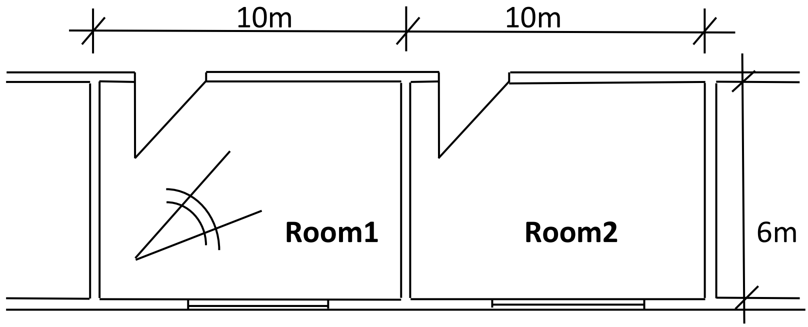
وفي ختام هذا الجزء نقول إن الأمثلة القادمة تخدم بالدرجة الأولى مهندسين قسمي مدني وعمارة ، حيث أننا نتطرق لعزل الفراغات المعمارية صوتيا ، هادفين بذلك إلى الإرتقاء بالمجتمع لمحيط أكثر هدوء، هذا الأمر سيجعلنا ننفصل عن السرد الذي كنا قد بدأناه في طبيعة الموجة الصوتية وأنواعها وما هو الثلاثي الأبعاد منها ، الأمر الذي سيجعلنا نفرد مؤلفا خاصا للموجات الصوتية.

نسأل الله أن يمد في أعمارنا حتى ذلك الحين ، وأن يجعل ما نقدمه صدقة جارية تنير القبور وتهدي على الصراط يوم لا ينفع مال ولا بنون إلا من أتى الله بقلب سليم.

Room Acoustic Calculations

Problem 2.1

Given are two spaces with the same dimension and the same conditions. Please evaleuate the sound transmission between the two rooms.



Given

Height of the room 3m

Sound power 1 75dB

Reverberation time 1.0 s

Difference between the two sounds power Δd = 42dB

Calculate

- The sound power in the second room
- The sound resistance in the wall between the two spaces
- We need to reduce the sound power in space 2 for about 15dB. Can we achieve it?

Solution

To calculate to sound power

$$Dn = L1 – L2 + 10\log \frac{A0}{A2}$$

With A0 = $10m^2$

$$L2 = L1 – Dn + 10\log\frac{10}{A2}$$

And A2 = $\frac{0.163}{T}$. V = $\frac{0.163}{1}$ .10.6.3 = $29{,}34m^2$

L2 = 75 – 42 +10log$\frac{10}{29{,}34}$ = 28,3dB

From the equations …

$$L2 = 20\log \frac{P2}{P0}$$

And P0 = 2.10^{-5}

$$P2= P0.10^{\frac{L2}{20}} = 2.10^{-5}.\ 10^{\frac{28{,}3}{20}} = 5{,}2.10^{-4}\ \text{Pa}$$

2- <u>Sound resistance</u>

$$R = L1 – L2 + 10\log \frac{S}{A2}$$

S = area of the separation wall, A = effective absorb area

R = 75 -28,3 + 10log $\frac{18}{29{,}34}$ = 44,6 dB

3- Sound power difference ΔL

In the ideal case αs for the wall equal 1

$A2 = A_f + A_W + A_c$

Which A_f = Area flooring

A_W = Area wall

A_c = Area Ceiling

$A2 = (2.10\ .6 + 2.6.3 + 2.10.3).\ \alpha s = 216m2$

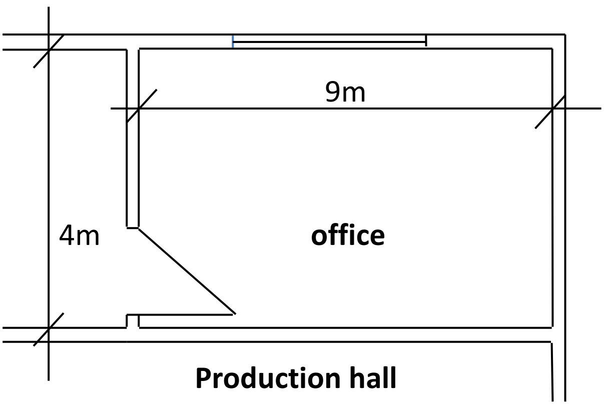
$\Delta L = 10\log\frac{29{,}34}{216} = -8{,}7\ dB$

<u>It means that the reduction from 15dB is not possible</u>

نستفيد من هذا المثال امكانية معرفة مدى تقليل الضوضاء الموجودة في فراغ ما بناء على المعطيات المتوفرة في المكان. يضاف إلى هذا حساب مقاومة الجدار الفاصل بين فراغين.

<u>Problem 2.2</u>

In a factory one administration office is separated from the production hall through a wall (Like the shown sketch)



Given

Room height 3,5m

The wall in between thickness 20cm

density 700kg/m3

elastic coefficient 2500 MN/m2

Sound power in the production hall 85dB

Sound power in office 45dB

Reverberation time 1s

Calculate

- Calculate the sound power and the sound intensity
- Is this wall, between the two spaces acoustical enough?

Solution

The norm sound power difference is:

$$Dn = D + 10 \log \frac{A0}{A} \quad dB$$

With…. A0 = 10 m2

And $A = 0.163 \frac{V}{T} = 0.163 \frac{9.4.3,5}{1,5} = 13{,}69m2$

Dn = 85 - 45 = 40 dB

Then $Dn = 40 + 10\log\frac{10}{13,69} = 38,6dB$

On the other hand..

$$\boxed{L = 10\log\frac{I}{I0} \quad = \quad 20\log\frac{P}{P0}}$$

And

$P_0 = 2.10^{-5}$pa , $I_0 = 10^{-12}$ w/m2

$P = P0.\ 10^{\frac{L}{20}} = = 2.10^{-5}.\ 10^{\frac{45}{20}} = 3,56.10^{-3}$ pa

Also the sound intensity

$I = I_0.\ 10^{\frac{l}{10}} = 3,16\ .10^{-8}$ w/m2

2) if the frequency < 1500Hz then it is not enough for achiving a suitable space concerning Sound.

$$\boxed{F_c = 64\ .\ \frac{1}{d}\sqrt{\frac{\rho}{E}}}$$

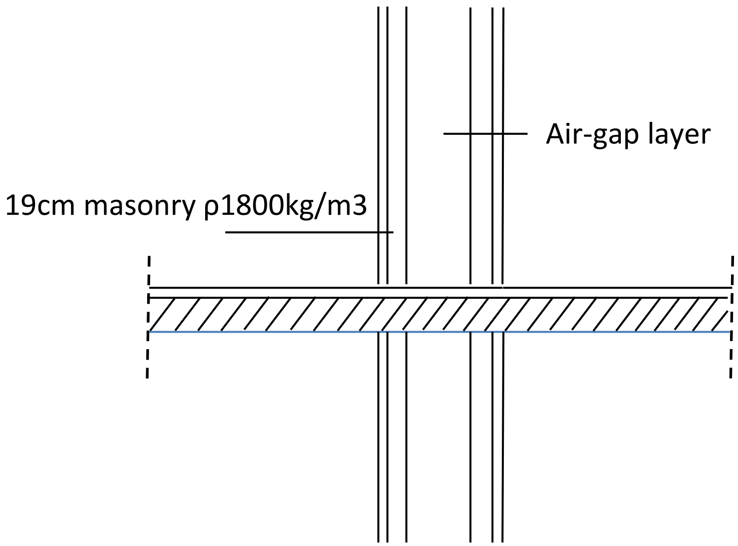
$F_c = 64\ .\ \frac{1}{0,2}\sqrt{\frac{700}{2500}} = 169$ Hz

علينا أن نعلم أننا في مقدورنا إختبار المواد في إمكانيتها للعزل الصوتي بتطبيق معادلة التردد الرنيني ، فإذا كان تردد الموجة الصوتية داخل

المادة خرسانة كانت أم طوب أحمر أم جبس أعلى من 1500 هيرتز كانت المادة عازل جيد للصوت . فإن كانت غير ذلك لم يكن عندها القدرة أو الكفاءة على خلق الفراغ الميح سمعيا.

Problem 2.3

Two houses are separated through two layering wall. The dimension are given through the following sketch.

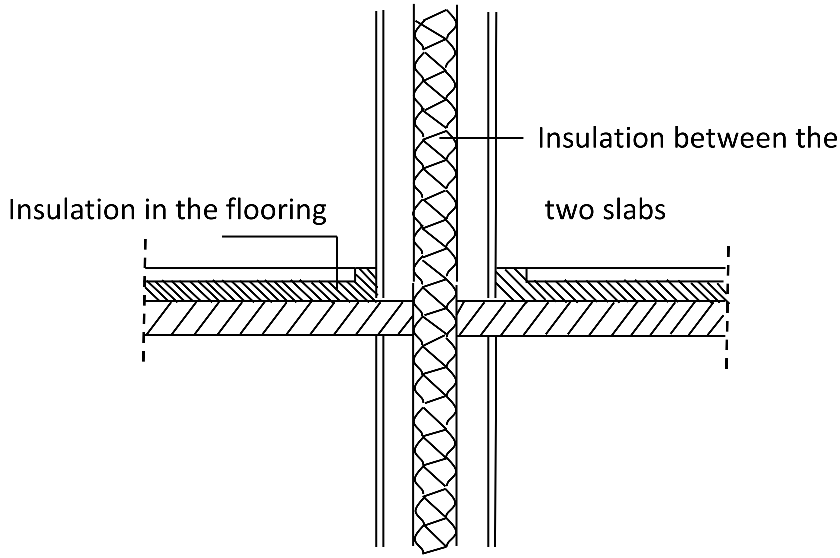


Calculate

- Evaluate the separation wall concerning acoustics
- If we have to correct the design of the wall, which criteries are to follow up
- How much is the thickness in the air-gap layer

Solution

- The construction is non-suitable concerning acoustics.
- The suitable solution is through separating of the flooring in the two units
- And through spreding the insulation in x- and y direction



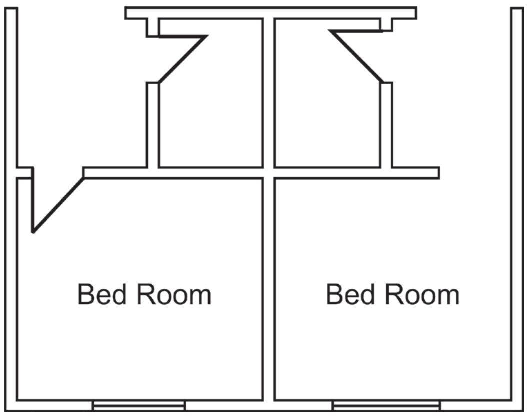
Only through this type of insulation is the sound isolated.

إن إستمرار البلاطة الخرسانية أو السقف بين فراغين يؤدي إلى جودة النقل للموجة الصوتية . وعليه فإن أي تصميم جيد للفراغ ينبغي أن يتجنب إتصال البلاطات بالإضافة إلى عزل الحوائط.

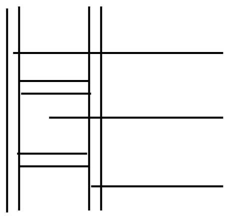
لكننا إنشائيا يصعب علينا ذلك ، فنحن لانستطيع بناء الفراغات منفصلة خاصة إذا كان البحر الذي نتحدث فيه بعد صغير ، حينها يكون البديل وضع العزل داخل البلاطة، أم الإكتفاء بالأركان حيث مسار الموجة الصوتية المتوقع.

Problem 2.4

Two units are back to back designed. The dimensions are given through the following sketch.



Height of the room 2,5m



1.5cm cement ρ = 1400kg/m3

24cm masonry ρ = 1800 kg/m3

1.5cm cement ρ = 1400 kg/m3

Air density 1.25kg/m3

Sound speed 340 m/s

Sound power L1 = 83dB, reverberation time 0,5 s, sound angle 45 , frequency 250 Hz

Calculate

- In the other room we must have a sound power of maximum 20dB. Please calculate the absorb area to keep the sound power 20dB

Solution

The sound power in the actual situation

$$R = 20\log \frac{\pi.f.m.\cos\delta}{\rho.c}$$

M = $\sum(\rho.d)$ = 1800. 0,24 + 2. 1400.0,015= 474 kg/m2

R = 20log() = 55,8dB $\frac{\sqrt{2}}{2}.\frac{\pi.250.474}{1.25.340}$

<u>The Sound Isolation</u>

R = L1-L2+10log $\frac{S}{A}$

With the area of the wall between the two spaces

S = 3 x 2.5 = 7.5m2

And the effective absorb area

A = 0.163 . $\frac{V}{A}$ = 0.163x 4x3x $\frac{2.5}{0.5}$ = 9.78m2

$\longrightarrow$ L2 = L1-R+10log $\frac{S}{A}$ = 83 -55.8+ 10log $\frac{7.5}{9.78}$ = 26dB

Therefore the separation wall between the two units is not enough to isolate the room

The needed absorb area

$R = L1 - L2 + 10\log\frac{S}{A}$ [dB]

$10\log\frac{S}{A} = R + L2 - L1 = 55.8 + 20 - 83 = -7.2$ dB

$A = S.\ 10^{0,72} = 7.5x10^{0,72} = 39.4m2$

It means we need 39.4m2 absorb area

إستطعنا في هذا المثال حساب المسطحات اللازمة بإستخدام قانون Sabine . طريقة سهلة في الحساب لكنها ليست الوحيدة.

Problem 2.5

A music room changed its function in a lecture hall. Please calculate the following.

Calculate

- The acoustic absorb area in the two cases
- The relation between the two sound intensity

Given

Room dimension 10 x 5 x 3 = 150m2

Solution

$$150m2 < 300m2$$

<u>According the European code</u>

The reverberation time for music T = 1,0 s

Specking T = 0,5 s

The absorb area A = 0,163 . $\frac{V}{T}$ [m2]

Music A = 0,163 . $\frac{150}{1,0}$ = 24,45m2

Speaking A = 0,163 . $\frac{150}{0,5}$ = 48,9m2

The relation between the two sound intensity

$\Delta L = 10 \log \frac{I2}{I0} - 10 \log \frac{I1}{I0} = 10 \log \frac{I2}{I1} = 10 \log \frac{A\ before}{A\ after}$

$= 10 \log \frac{24,45}{48,9} = 10 \log 0,5$

$10 \log \frac{I2}{I1} = 10 \log 0,5$

I2 = 0,5 I1

Problem 2.6

To reduce the reverberation time of one room with a volume from 150m3, we have to use absorb wall from25m2. Please calculate the following.

Calculate

- Which thickness must has the absorb wall, if we have in this room a frequency from 3000Hz must be absorbed to have air temperature from 15̊c.
- The absorb wall include an absorption grad from 0.9 and its absorbed ten times than without absorpation. what is the relation between the two reverberation times T1 and T2
- What is the maximum and minimum of the reverberation time in this room?

Given

Air adiabatic exponent 1.4

Gas constant 290 Ws/kgk

Solution

The thickness of the absorb wall is $\frac{\lambda}{4}$[m]

d= $\frac{\lambda}{4}$ [m] ; λ = $\frac{c}{f}$ [m] ; c = $\sqrt{k.R.t}$ = $\sqrt{1.4\ x\ 290\ x\ (273+15)}$

= 342 m/s

d = $\frac{c}{4.f}$ = $\frac{342}{4.3000}$ = 0,03 m = 3cm

b) The reverberation time

$$T = 0{,}163 \cdot \frac{V}{A}\ [s]$$

The absorptions grad: $\alpha1= 0.09$,

$\alpha2 = 0.9$

Area 1 = 25 x 0.09 =2,25m2

Area 2 = 25 x 0.9 = 22,5m2

So the reverberation time is:

$T1 = 0.163 \cdot \frac{V}{2,25+AR}$

$T2 = 0.163 \cdot \frac{V}{22.5+AR}$

Note that A_R is the area around the sound source. $A_{R=0}$ by max reverbation time

$2.25 = 0.163 \frac{V}{T1}$

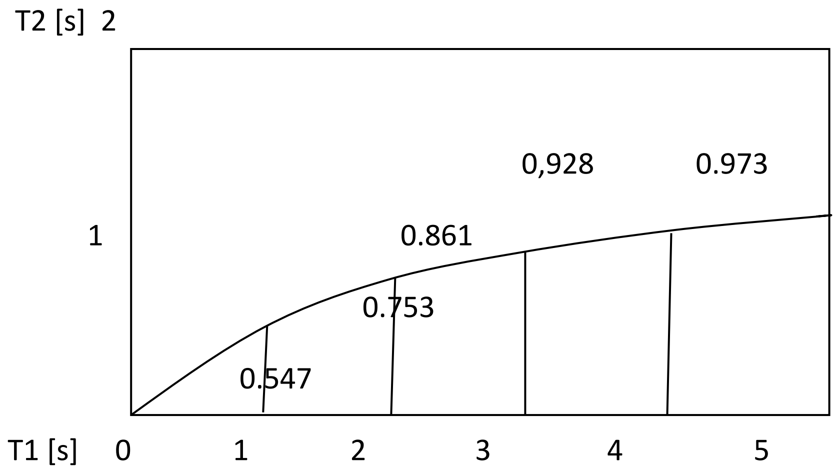
$22.5 = 0.163 \frac{V}{T2}$

$2.25 - 22.5 = 0.163 \cdot V \cdot \left(\frac{1}{T1} - \frac{1}{T2}\right)$

$\frac{1}{T1} - \frac{1}{T2} = \frac{2.25-22.5}{0.163 \ x\ 150} = -0.828$

$\frac{1}{T2} = \frac{1}{T1} + 0.828$

$T2 = \frac{1}{\frac{1}{T_1}+0.828}$ [s]



c) The maximum reverberation time is in case $A_R = 0$

T1 max = $\frac{0.163 \, x \, 150}{25 \, x \, 0.09}$ = 10.9 s

T2 max = $\frac{T1}{10}$ = 1.09 s

استطعنا في هذا المثال معرفة سمك الطبقة العازلة واستطعنا أيضا حساب زمن الصدى المحتمل في حالته القصوى وحالته الدنيا ، مع العلم أن زيادة المسطحات الماصة وشبه الماصة في أي فراغ تقلل من زمن الصدى. ودليل ذلك أن الغرفة الخالية تسمع فيها صدى الصوت بوضوح أما الغرفة المفروشة فيتلاشى أو ينعدم زمن الصدى.

ولاريب أن العلاقة شبه طرديه بين الزمن الأقصى والزمن الأدنى ، فزيادة قيمة الأول يؤثر إيجابا على الزمن الثاني.

وهنا يظهر سؤال في أي الفراغات يكون الإحتياج إلى انعدام الصدى في أقصى مراتبه، والإجابة بلا شك فراغات المسارح وقاعات الإحتفالات الكبرى ودور السينما . وعلى من يتصدى للعملية التصميمية أن يكون

على دراية بعلوم نقل الصوت وكيفية عزلها والتعامل معها. ولا يفوتني أن أقول إن ما نتطرق إليه بالحديث اليوم ليس علما حديثا ولكن عرفته البشرية منذ الآف السنين وأدلى فيه المسلمون بدلوهم لذا كان تصميم المحراب داخل المساجد على شكل نصف كره ، حتى يتوزع صوت الإمام على رواد الصف الأول ومن يليهم من الصفوف بالتساوي.

Problem 2.7

What is the relation between the following items?

- Sound pressure and absorption area
- Sound intensity and absorption area
- Sound pressure and sound intensity

Solution

From the equation ΔL

$$\Delta L = 10\log\frac{A1}{A2} = 20\log\frac{P2}{P1} \quad [dB]$$

$$20\log\frac{P2}{P1} = 10\log\frac{A1}{A2}$$

$$\frac{P2}{P1} = \left(\frac{A1}{A2}\right)^{\frac{1}{2}}$$

$$P2 = P1 \cdot \sqrt{\frac{A1}{A2}} \quad [Pa]$$

b) In the same way

$$\Delta L = 10 \log\frac{A1}{A2} = 10 \log \frac{I2}{I1} \text{ [dB]}$$

$$10 \log \frac{I2}{I1} = 10 \log\frac{A1}{A2}$$

$$I2 = I1 \cdot \frac{A1}{A2}$$

c) from the same relation

$$10 \log \frac{I2}{I1} = 10 \log \left(\frac{P2}{P1}\right)^2$$

$$P2 = P1 \sqrt{\frac{I2}{I1}} \text{ [Pa]}$$

<u>Problem 2.8</u>

One space must be diveided in two spaces. The surface of the walls has the following data:

Room dimension 8.0 x 5.0 x 2.5m

Walls cement $\alpha s = 0.02$

Flooring $\alpha s = 0.05$

Ceiling $\alpha s = 1.0$

Separating walls dimension 5.0 x 2.5m

Calculate

- The reverberation time before separating
- The thickness of the separation wall if the density of the wall 1800 kg/m3 to have the acoustic resistance of 54 dB
- Δ L between the two spaces , if the separating wall has the acoustic resistance of 50 dB

Solution

T = 0,163 . $\frac{V}{A}$ [s]

V = 5 x 8 x 2.5 = 100m3

A = $\sum$ (si . αi) = 2 x 2.5 x 8 x 0.02 + 2 x 2.5 x 5 x 0.02 + 5 x 8 x 0.05 + 5 x 8 x 1.0 = 43.3m2

T = 0.163 $\frac{100}{43.3}$ 0.38 s

The reverberation time is suitable

b- the thickness of the separation wall

Rw = 38 + 26.7 log $\frac{m}{100}$

$\frac{54-38}{26.7}$ = log $\frac{m}{100}$

m = 100 x $10^{0.6}$ = 398.1 kg/m2

d = $\frac{m}{\rho}$ = $\frac{398.1}{1800}$ = 0.22m = 22cm

Δ L = R – 10 log $\frac{S}{A}$

S = 5 x 2.5 = 12.5m2

Length of the new separated room: 0.5 (8 – 0.24) = 3.88m

A = $\sum$ (Si.αi) =

= (5 + 2 x 3.88) . 2,5 x 0.02 + 5 . 3,88 . 0,05 + 5 . 3,88 . 1,0

+ 5 . 2,5 . 1,0 = 33,5 m2

D = 50 – 10 log $\frac{12,5}{33,5}$ = 54,3dB

Problem 2.9

In a small factory three machines are exsisting in room 1, and in room 2 we found only one machine. Otherwise we have furniture and absorb areas from 2.5m2

Given

Room dimension 20 x 20 x 5

Ceiling αs = 0.09

Flooring αs = 0.04

Walls αs = 0.01

Sound intensity in room 1 3,2 x 10^{-4} w/m2

in room 2 1.0 x 10^{-4} w/m2

In the questions 1-3 you have to neglect the sound intensity through the separating wall.

Calculate

- The reverberation time in the two rooms
- The sound power L in the two rooms
- In room 1 we have to reduse 5dB. Which material can we use on the wall as absorb material
- Which resistance must the separating wall had, to have 60dB in room 2

Solution

The reverberation time

$T = 0.163 \frac{V}{A} = 0.163 \frac{20 .20.5}{A}$ [s]

Room1

A1 = A + 2,5m2

A = ∑ (Si.αi) = 20 . 20 . (0,09 + 0,04) + 4 . 20 . 5 . 0,01 = 56m2

A1 = 56m2 + 2,5m2 = 58,5m2

$T1 = 0,163 . \frac{2000}{58.5} = 5.6$ s

Room2

A2 = A = 56m2

$T2 = 0,163 \frac{2000}{56} = 5,8$ s

Sound power L

L = 10 log $\frac{I}{Io}$, with Io = 10^{-12}

We have 3 machines in room 1

L1 = 10 log $\frac{3x\ 3{,}2 . 10^{-4}}{10^{-4}}$ = 90dB

L2 = 10 log $\frac{10^{-4}}{10^{-12}}$ = 80dB

Sound power difference

ΔL = 10 log $\frac{A\ before}{A\ after}$ = -5dB

A $_{before}$ = 58,5m2

A $_{after}$ = $10^{0,5}$. A $_{before}$ = 185m2

A $_{after}$ = 20 x 20 (0,09 + 0,04) + Aw + 2,5m2 = 185m2

Aw = 185 – 52 – 2,5 = 130,5m2

Aw = αw . sw

If the area of wall is

Sw = 4 x 20 x 5 = 400 m2

αs = $\frac{130{,}5}{400}$ = 0,33 that can be wood recovering with 20mm thickness

The resistance of the separating wall

R = D + 10 log $\frac{S}{A2}$ [dB]

$D = L1 – L2 = 85 – 60 = 25dB$

$S = 20 . 5 = 100$ m2

$R = 25 + 10 \log \frac{100}{56} = 27,5$ dB

<u>Problem 2.10</u>

Two Conference halls have a separating wall. The sound power in one hall is 72 dB.

Given

Conference hall dimension

6,5 x 5,0 x 3,0 m

Separating wall

5,0 x 3,0m

Density of the wall 650 kg/m3

Density of the air 1,25 kg/m3

Sound speed 343 m/s , sound angle 45°, Frequency 100Hz

Calculate

- The absorb area to have ideal acoustic conditions
- The thickness of the separating wall to have sound power of 35dB
- To have better acoustics in the room, we must build a gypsum board panel with a thickness of 12,5mm and the density of

900kg/m3. Which reverberation frequency are we have if the airgap between the two walls is 5cm

Solution

Absorb area

A = 0,163 . $\frac{V}{T}$ [m2]

V = 6,5 x 5 x 3 = 97,5 m3

A = 0,163 $\frac{97,5}{0,5}$ = 31,8 m2

The thickness of the separating wall

R = D + 10 log $\frac{S}{A}$ [dB]

S = 5 x 3 = 15m2

R = 37 + 10 log $\frac{15}{31,5}$ = 33,7 dB

R = 20 log $\frac{\pi . f . m . \cos v}{\rho . c}$ = 20 log $\frac{\pi . 100 . \rho . d . \sqrt{2}}{1,25 . 343 . 2}$ = 33.7dB

The minds thickness d ≥ $\frac{1,25 . 343 \sqrt{2}}{\pi . 100 . 650}$. $10^{\frac{33,7}{20}}$ ≥ 0,14m

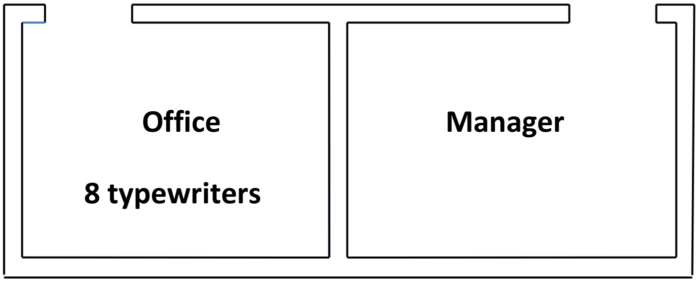
The reverberation frequency

fr = $\frac{60}{\sqrt{m . d}}$ = $\frac{60}{\sqrt{900 . 0,0125 . 0,05}}$ = 80 Hz

This is a suitable construction because 80Hz ≤ 100Hz (the frequency in the space)

Problem 2.11

One separating wall between two spaces has the resistance of 40dB.



Given

Office dimension

10,0 x 4,0 x 3,0 m

Manager

4,0 x 4,0 x 3,0 m

Reverberation time in manager office 0,5 s

Typewriters per uni 10^{-5} w/m2

Separating wall density 1000kg/m3

Air density 1,25kg/m3

Sound speed 343m/s Sound angle 45°

Calculate

- The total sound power from the typewriters
- The thickness of the separating wall
- By which frequency is the resistance of the separating wall equal with the resistance of the building
- Is it enough, the resistance of the separating wall to have 35dB in the manger office.

Solution

L total = L1 + ΔL [dB]

L1 = 10 log $\frac{I}{Io}$ = 10 log $\frac{10^{-5}}{10^{-12}}$ = 10 log 10^{7} = 70 dB ;

Io =10^{-12}w/m2

ΔL = 10 log n = 10 log 8 = 9 dB

L total = 70 dB + 9 dB = 79 dB

Thickness of the wall

$$\boxed{d = \frac{m}{\rho}}$$

Rw = 38 + 26,7. log$\frac{m}{100}$ = 40dB

m = 100. $10^{\frac{2}{26,7}}$ = 118,8 kg/m2

d = $\frac{118,8}{1000}$ = 0,12m

Frequency

Under the suggestion that

R = 2olog $\frac{\pi . f . m . \cos v}{\rho . c}$ = Rw = 40dB

f = $\frac{\rho . c}{\pi . m . \cos v} . 10^{\frac{R}{20}} = \frac{1{,}25 . 343}{\pi . 118{,}8} \sqrt{2} . 10^{2}$ =162.5 Hz

The resistance what we need

Dneed = 79dB – 35dB = 44dB

Dexist = R - 10log $\frac{S}{A}$ [dB]

S = 3 x 4 = 12 m2

A = 0,163 $\frac{V}{A}$ = 0,163 . $\frac{4 . 4 . 3}{0{,}5}$ = 15,65m2

Dexist = 40 – 10log $\frac{12}{15{,}65}$ = 41,2dB

Rneed = Dneed + 10log $\frac{12}{15{,}65}$ = 44 + (-1,2) = 42,8dB

<u>لابد لنا أن نعلم أننا في موجات الصوت نفرق بين التالي:</u>

-الجدران الأحادية One layering ولها المعادلات التالية

$$f = 64 \frac{1}{d} \sqrt{\frac{\rho}{E}}$$

$$Rw = 37,5 \cdot \log m - 42$$

$$Rw = 38 + 26,7 \log \frac{m}{100}$$

-الجدران الثنائية Two layering ولها المعادلات التالية

$$R = R_1 + R_2 + 20\log \frac{4\pi f d}{c}$$

Only if.... $f_R < f < \frac{c}{4.d}$

$$R = R_1 + R_2 + 6$$

if.... $f > \frac{c}{4.d}$

$$f_R = \frac{1}{2\pi}\sqrt{s\left(\frac{1}{m1} + \frac{1}{m2}\right)}$$

Note that

m1 = mass of related area of layer1 [kg/m2]

m2 = mass of related area of layer2 [kg/m2]

S = Stiffness of the material [N/m3]

f = frequence [Hz]

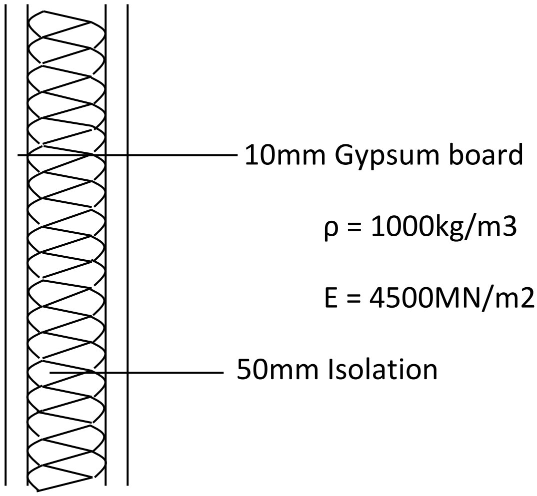
f_R = Resonance frequence [Hz]

مـع العلـم أن التـردد الرنينـي f_R لا يحـدث إلا فـي الجـدران الثنائيـة. وبالنسبة لنا المادة المرنة الماصة للموجـات الصـوتية الجيـدة مـن حيـث العزل هي التي يزيد ترددها عن 1500 هيرتز.

وفـي المثـال رقـم 12 بالطبقـة النحيفـة والطبقـة غيـر العازلـة بالطبقـة السميكة –راجع المثال-

Problem 2.12

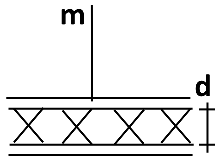
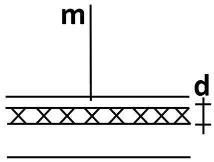
Check the separating wall with the following given data.



Calculate

- Sketch the relation between the frequency and the construction type of the wall
- Evaluate the construction system of the wall

Solution

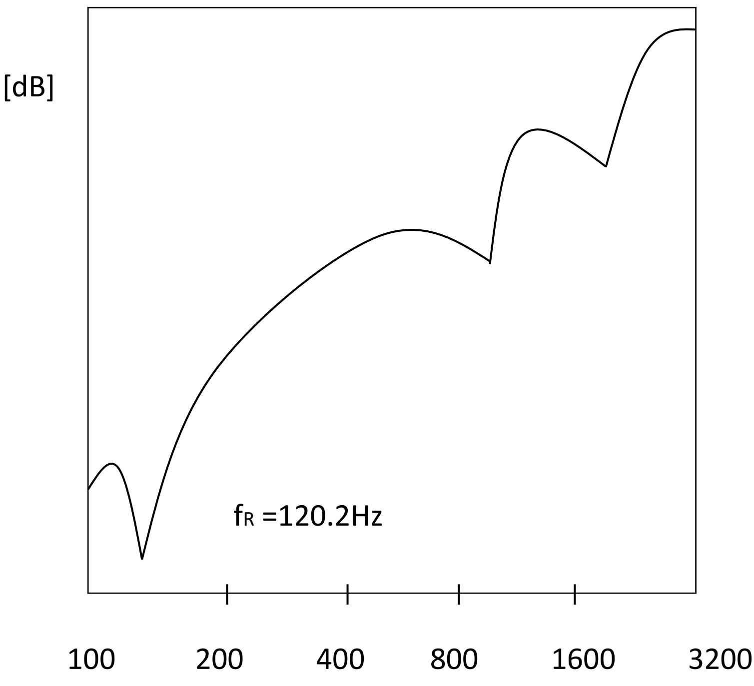
Room in between		
Air-gap Layer	$f_R = \frac{85}{\sqrt{m.d}}$	$f_R = \frac{60}{\sqrt{m.d}}$
Insolation layer	$f_R = 225\sqrt{\frac{s}{m}}$	$f_R = 160\sqrt{\frac{s}{m}}$

<u>Note that…</u>

S = Stiffness of the material [N/m3]

m = Mass of the solid layer [kg/m2]

d = Thickness of the Isolation layer [m]



$f_R = \frac{85}{\sqrt{m.d}} = \frac{85}{\sqrt{10\ x\ 0.05}}$ = 120.2 Hz

m = $\boldsymbol{\rho}$. d = 1000 . 0,01 = 10 kg/m2

The frequency is unsuitable because the frequency of the wall is in the zone of the frequency of the space.

Problem 2.13

One seminar room must be suitable concerning acoustics. Please use the given data to calculate the following.

Given data

Reverberation time	0,8s
Volume	120m3
Adiabatic constant	1.4
Gas constant	290ws/kgk

Calculate

- The absorption area in the room
- The reverberation time to have suitable acoustics
- The distance between the recovering material and the wall, if the temperature 20°c and the frequency of the surface more than 2kHz

Solution

The absorb area

A = 0,163 . $\frac{V}{T}$ = 0,163 . $\frac{120}{0,8}$ = 24,45m2

Because the volume of the space is to small $<$ 300m2, then the reverberation time must be 0,5 s

The distance is to calculate through the equation

$$d = \frac{\lambda}{4}$$

and λ = $\frac{c}{f}$; and c = $\sqrt{k\,.R\,.T}$ = $\sqrt{1{,}4\,.\;290\,.(273+20)}$ =345m/s

λ = $\frac{345}{2000}$ = 0,173 m

d ≥ $\frac{0{,}173}{4}$ = 0,043 m

Problem 2.14

Outside wall, 60% masonry and 40% glass must be acoustic investigated. Please use the given data to calculate the following.

Given data

Outside wall 24cm

Density 800kg/m3

Water part 30%

Glass acoustic resistance 40dB

Calculate

- Acoustic resistance from the masonry wall
- After drying the water part will be only 4%. Please calculate:
- The mass of the water pro m2
- How many dB is the wall better after drying
- The resistance of the total area
- Which treatment is available for the total area

Solution

The equation of homogenous wall one layer is:

$$R_w = 37{,}5 \cdot \log m - 42 \quad [dB]$$

The mass of the wall exist from the drying part and the part with water

$$m = m_{dr} + m_w$$

$\rho_{dr} \cdot d_{dr} + u_v \cdot \rho_w \cdot d_w = 800 \cdot 0{,}24 + 0{,}3 \cdot 1000 \cdot 0{,}24 = 264 kg/m2$

$$Rw = 37{,}5 \cdot \log 264 - 42 = 48{,}8 \text{ dB}$$

<u>The mass of the water pro m2</u>

$Mw = 0{,}04 \cdot \rho_w \cdot d_w = 0{,}04 \cdot 1000 \cdot 0{,}24 = 9{,}6 kg/m2$

The dB after drying

$m = m_{dr} + m_w = 800 . 0{,}24 + 9{,}6 = 201{,}6$ kg/m2

$R_{w2} = 37{,}5 . \log 201{,}6 - 42 = 44{,}4$ dB

$\Delta R = R1 - R2 = 48{,}8 - 44{,}4 = 4{,}4$ dB

The total resistance

$$R_{total} = R2 - 10\log\left[1 + \frac{total\ area}{glass\ area}\left(10^{\frac{R_2 - glass\ part}{10}} - 1\right)\right]\ [dB]$$

Total area = 7 . 3,5 = 24,5m2

Glass area = 7 . 3,5 . 0,4 = 9,8 m2

$$R_{total} = 44{,}4 - 10\log\left[1 + \frac{9{,}8}{24{,}5}\left(10^{\frac{44{,}4-40}{10}} - 1\right)\right] = 42{,}1\ dB$$

The available treatment is

- Wall recovering
- Better glass

Problem 2.15

One wall consist only homogenous building material. Please calculate the following.

Given

Light Concrete Thickness 24cm

Elastic module 3800 MN/m2

Density 1300 kg/m3

Calculate

- The speed of the acoustic waves in the wall
- The frequency in the wall
- The resistance of the wall
- We build a recovering wall from wood with the masse of 10kg/m2. Which resonance frequency are we have if the room in between 20mm
- Evaluate the wall from the side of the acoustic behavior

Solution

The speed of the acoustic waves (longitude)

$$C_L = \sqrt{\frac{E}{\rho} \cdot \frac{1-v}{(1+v)(1-2v)}}$$

$$C_L = \sqrt{\frac{3800 .10^6}{1300} \cdot \frac{1-0,15}{(1+0,15)(1-20,15)}} = 1757 m/s$$

The speed of the acoustic waves (Transvers)

$$C_{Tr} = \sqrt{\frac{E}{\rho} \cdot \frac{1}{2(1+v)}}$$

$$C_{Tr} = \sqrt{\frac{3800 .10^6}{1300} \cdot \frac{1}{2(1+0,15)}} = 1127 m/s$$

The coincidence frequency in the wall

$$f = 64 \cdot \frac{1}{d} \cdot \sqrt{\frac{\rho}{E}} = 64 \cdot \frac{1}{0,24} \cdot \sqrt{\frac{1300}{3800}} = 156 \text{ Hz}$$

The resistance of the wall

R = 37,5 . log m - 42dB

m = ρ . d = 1300 . 0,24 = 312 kg/m2

R = 37,5 . log 312 – 42 = 51,5 dB

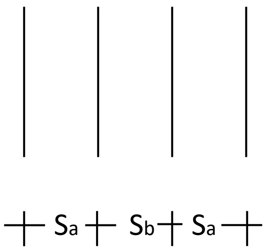
Resonance frequency

$$f_r = \frac{60}{\sqrt{m.d}} = \frac{60}{\sqrt{10 .0,02}} = 134 \text{ Hz}$$

the construction system is non-suitable because fr > 100 Hz

Problem 2.16

The wall construction is like the following sketch



	Layer A	Layer B	Total construction
Thickness	-	-	21
Density	700	300	-
Elastic module	$3,8.10^{9}$	3.10^{6}	-
Temperature ability	0,22	0,04	-
Heat transfer coefficient	-	-	0,41

Calculate

- Which material can we use by this construction
- The thickness of the wall and the coincidence frequency
- The total coincidence frequency for the wall
- Sketch for the total wall the acoustic resistance
- Evaluate the wall from the side of the acoustic behavior

Solution

The material concerning data is

Layer A : concrete

Layer B : Isolation

The thickness of the layers

$$K = [\frac{1}{\alpha i} + \frac{s1}{\lambda 1} + \frac{s2}{\lambda 2} + \frac{s3}{\lambda 3} + \frac{1}{\alpha o}]^{-1} = 0,41 \text{ w/m2k}$$

$$2 . \frac{Sa}{\lambda a} + \frac{Sb}{\lambda b} = \frac{1}{0,41} - \frac{1}{\alpha i} - \frac{1}{\alpha o} = \frac{1}{0,41} - 0,13 - 0.04 = 2,27 \text{ m2k/w}$$

Also

2.Sa + Sb = 0,21 m ; 2.Sa = 0,21 – Sb

$$Sb = \frac{2,27 - \frac{0,21}{\lambda a}}{\frac{1}{\lambda b} - \frac{1}{\lambda a}} \quad ; \quad Sa = 7,5\text{cm} \quad ; Sb = 6\text{cm}$$

The coincidence frequency

$$f_c = 64 . \frac{1}{d}\sqrt{\frac{\rho}{E}} = 64 . \frac{1}{0,075} . \sqrt{\frac{700}{3800}} = 366 \text{ Hz}$$

The layer A is isolation effective because fc = 366 Hz < 1500 Hz

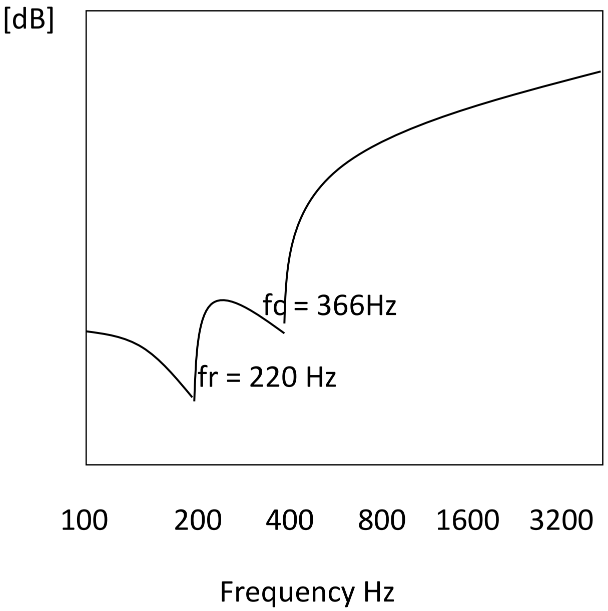
The resonance frequency

$$f_R = \frac{1}{2\pi} . \sqrt{s . (\frac{1}{m1} + \frac{1}{m2})}$$

$$Sb = \frac{E}{d} = \frac{3.10^6}{0,06} = 50 \text{ MN/m3}$$

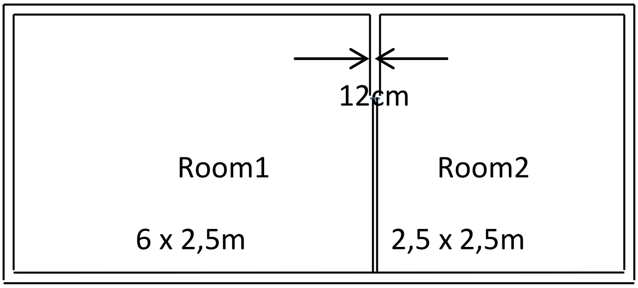
m1 = m2 = mA = 700 . 0,075 = 52,5 kg/m2

$$f_R = \frac{1}{2\pi} \cdot \sqrt{50 \cdot 10^6 \cdot \frac{2}{52{,}2}} = 220 \text{ Hz}$$



Problem 2.17

Two rooms with the following dimension



Calculate

- The reverberation time in the two rooms
- The absorption areas in the two rooms
- The resistance of the separation wall is 42dB and the door content only 37dB resistance. Please calculate the resistance of the wall
- The sound intensity in room B, if the intensity in room A 20dB, and the diffuse of the door 60dB
- Through which way is the transfer of the sound waves

Solution

The ideal reverberation time for spaces ≤ 300m3 is o,5 s

V1 = 6 x 4,8 x 2,5 = 72m3

V2 = 2,5 x 4,8 2,5 = 30m3

So we will adapt the reverberation time of 0,5 s

Then

A1 = 0,163 $\frac{72}{0,5}$ = 23,5m2

A2 = 0,163 $\frac{30}{0,5}$ = 9,8m2

The total resistance of any element with different resistance is through the following equation:

R total = R wall – 10 log [1 + $\frac{S\ door}{S\ total}$ ($10^{\frac{R\ wall - R\ door}{10}}$ -1)]

S door = 2 x 1 = 2m2 , S total = 4,8 x 2,5 = 12m2 , R door = 37dB , R total = 42dB

$$R\ wall = 10\log \frac{1-\frac{S\ door}{S\ total}}{10^{\frac{-R\ total}{10}} - \frac{S\ door}{S\ total}.10^{\frac{-RT}{10}}}$$

$$R\ wall = 10\log \frac{1-\frac{2}{12}}{10^{\frac{-42}{10}} - \frac{2}{12}.10^{\frac{-37}{10}}} = 44dB$$

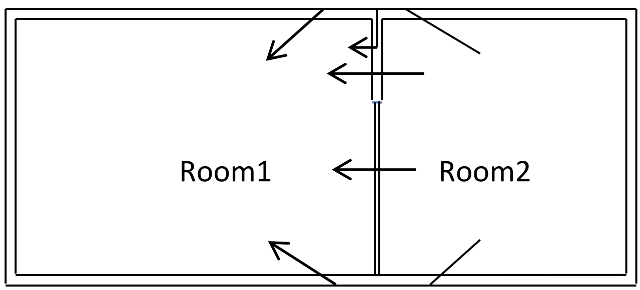
<u>The resistance of the wall</u>

$$R\ total = D + 10\log\frac{S\ total}{A\ 1} = 60\ dB$$

$$D = L2 - L1 = R - 10\log\frac{12}{23,5} = 63dB$$

L2 = L1 + 63dB = 20 + 63 = 83dB

The path of the sound wave transmission



Problem 2.18

In an industrial building we have two rooms with the same dimension. In one room exist two machines. The third one exist in the other room. The sound intensity for each machine $6,4.10^{-3}$w/m2. Please keep in mind that we have 4m2 wood recovering material as an absorption area in room2.

Given

Room dimension 15 x 15 x 4m

Sound absorption by 1000Hz

Ceiling	0,09
Flooring	0,04
Walls	0,01

Calculate

- The reverberation time in the two rooms
- The sound intensity if all machines are on
- Which absorption coefficient must we have in the louder Room to have the same intensity in the two rooms.

Solution

$T = 0{,}163\ \frac{V}{A}$

V = 15 x 15 x 4 = 900m2

A = (2.l.h + 2.w.h)αw + l.w. αc + l.w.αf =

= (2.2.15.4)0,01 + 15 .15.0,09+ 15.15.0,04 = 31,7m2

Room 1 A1 = A + 4m2 = 35,7m2

T1 = 0,163 $\frac{900}{35,7}$ = 4,1s

Room 2 A2 = A = 31,7s

T2 = 0,163 $\frac{900}{31,7}$ = 4,6 s

<u>Sound intensity</u>

L = 10log $\frac{I}{Io}$ [dB] ; Io = 10^{-12} w/m2

Room 1 L1 = 10log $\frac{2 \, x \, 6{,}4.10^{-8}}{10^{-12}}$ = 101 dB

Room 2 L2 =10log $\frac{6{,}4 .10^{-8}}{10^{-12}}$ = 98dB

<u>The difference between the two rooms</u>

ΔL = L1 – L2 = 101 – 98 = 3dB

To have the same intensity in both rooms

$$\Delta L = 10\log \frac{A \ after}{A \ before}$$

A after = $10^{0,3}$. A before = $10^{0,3}$. 35,7 = 71,2m2

In the same time

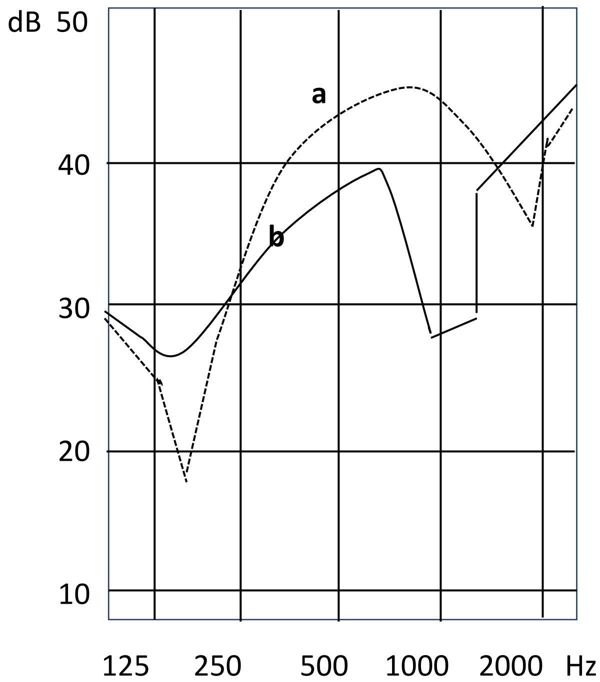
A after = 35 + 15 x 15 x (α – 0,9)

<u>The absorption coefficient</u>

$$A = \frac{A\ after - 35{,}7 + 15\ x\ 15\ x\ 0{,}9}{15\ x\ 15} = \frac{71{,}2 - 35{,}7 + 15x\ 15\ x\ 0{,}09}{15\ x15} = 0{,}25$$

Problem 2.19

The frequency in two buildings element is like the following diagram. One element is one layering and the other is double layering, with the same area for both of them.



Given

Separation wall 4 x 3m ; 30 kg/m2

One layer building element

Thickness 12mm

Air density 1,25 kg/m3

Air speed 345m/s

Calculate

- The critical frequency for that building elements
- Which curve in this diagram is for the double layering wall
- Evaluate the both walls concerning acoustics
- Which distances is suitable for the air layer in the doublelayering wall
- The elastic module for the one layering wall
- The sound intensity in the room if we have 1000Hz frequency and 0,11 N/m2 sound pressure and 20m2 absorption area

Solution

The critical frequency for that building element is :

Curve a : fc = 160 Hz and 2000 Hz

Curve b : fc = 1000 Hz

The curve a is double layering because of the raise of the curve.

The Evaluation

Both of the curve are unsuitable. We have in both of them a brokenness in the continuation of the curve.

One layering = 1000Hz

Dapple layering = 160Hz

The distance between the layers

$$fr = \frac{1}{2\pi} \cdot \sqrt{Sl\left(\frac{1}{m1} + \frac{1}{m2}\right)}$$

which s is the air dynamic constant

$$S = \frac{\rho . c^2}{d} = \frac{1{,}25 . 345^2}{d}$$

$$fr = \frac{1}{2\pi} \cdot \sqrt{\frac{1{,}25 . 345^2}{d} . 2 \frac{1}{m}}$$

$$fr = 86{,}8 . \sqrt{\frac{1}{m . d}} = 160 \text{ Hz}$$

the distance $d = \left(\frac{86{,}8}{160}\right)^2 . \frac{1}{m} = \frac{0{,}294}{15} = 0{,}02m$

The Elastic module

Because of the brokenness in the continuation of the curve by 1000Hz So

$F = 64 . \frac{1}{d} \sqrt{\frac{\rho}{E}} = 1000Hz$

$\rho = \frac{m}{d} = \frac{30}{0,012} = 2500kg/m3$

$E = (\frac{64}{d.f})^2 . \rho = (\frac{64}{0,012 .1000})^2 . 2500 = 71. 10^3$ MN/m2

This material can be glass

<u>The sound intensity</u>

$L2 = L1\text{-}R + 10\log\frac{S}{A}$; $S = 4 \times 3 = 12m2$

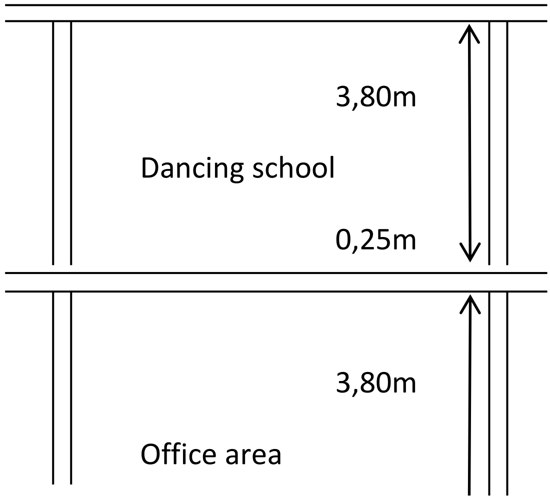
$L1 = 20\log\frac{p}{Po}$

$L1 = 20\log \frac{110.10^{-3}}{2.10^{-5}} = 75$ dB

$L2 = 75 - 44 + 10\log \frac{12}{20} = 29$ dB

<u>Problem 2.20</u>

A reinforced concrete slab separate a dancing school and an office area like the following section:



Given

Density of the slab 2400 Kg/m3

Elastic module 30.10^3 MN/m2

Office absorb area 40m2

Calculate

- The resistance of the ceiling
- The coincidence frequency of the ceiling
- The reverberation time in the office
- The absorption area in the dancing school to have ideal resistance.
- The sound intensity in the office area if you have sound pressure from 0,4N/m2
- The norm sound intensity

Solution

$R = 37{,}5 \cdot \log m - 42dB$

$m = \rho \cdot d = 2400 \cdot 0{,}25 = 600Kg/m2$

$R = 37{,}5 \cdot \log 600 – 42 = 62$ dB

The coincidence frequency of the ceiling

$$f = 64 \cdot \frac{1}{d}\sqrt{\frac{\rho}{E}} = 64 \cdot \frac{1}{0{,}25}\sqrt{\frac{2400}{30.10^8}} = 72 \text{ Hz}$$

The reverberation time

$T = 0{,}163\,\frac{V}{A}$; $V = 3{,}8 \times 5 \times 8 = 152m3$

$$T = 0{,}163\,\frac{150}{40} = 0{,}62 \text{ s}$$

The absorption area in the dancing school to have ideal resistance.

For a volume < 300m3 the ideal reverberation time is 1s

$$A = 0{,}163\,\frac{V}{T} = 0{,}163\,\frac{152}{1{,}0} = 24{,}8m2$$

The sound intensity

$L = 20\log\frac{P}{Po}$; $Po = 2.\ 10^{-5}$

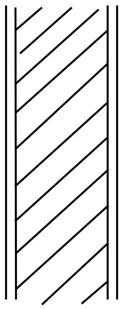
$L = 20\log\frac{0{,}4}{2.10^{-5}} = 86dB$

<u>The norm sound intensity</u>

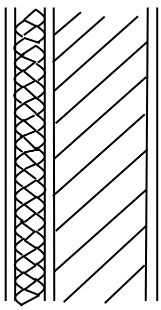
$Ln = L + 20\log\frac{A}{Ao} \quad = 86 + 10\log\frac{40}{10} = 92dB$

Problem 2.21

An outside wall must changed in a thermal resistances wall like the following sketch.



Before



After

Given

Wall A 1,5 cm gypsum board ρ = 1200 kg/m3

24 cm masonry ρ = 1200 kg/m3

2,0 cm gypsum board ρ = 1400 kg/m3

Wall B   1,5 cm gypsum board   ρ = 1200 kg/m3

24 cm masonry   ρ = 1200 kg/m3

2,0 cm gypsum board   ρ = 1400 kg/m3

5,0 cm Isolation   ρ = 30 kg/m3

1,5 cm light gypsum board   ρ = 1100 kg/m3

Air density   ρ = 1,25 kg/m3

Sound wave perpendicular 100Hz

Sound speed 330 m/s

Calculate

- The resistance of the wall A
- The speed in wall A, if the elastic module 16.10^{9} N/m2
- The dynamic stiffness coefficient of the isolation layer to have a resonance frequency of 80 Hz
- The resistance of the wall B
- The glass area if the resistance of the glass 35dB, and the resistance for the total façade is 40dB.
- The absorption grad of the inside wall B if L outside 70dB and L inside 22 dB .

Solution

The resistance of the wall A

R= 20log $\frac{\pi . f . m}{\rho . c}$. cosv

$M = \sum \rho \cdot d = 1200 \cdot 0{,}015 + 1200 \cdot 0{,}24 + 1400 \cdot 0{,}02 = 334 kg/m2$

$V = 0° \longrightarrow \cos v = 1$

$R = 20\log \frac{\pi \cdot 100 \cdot 334}{1{,}25 \cdot 330} = 48$ dB

The sound speed in wall A

$$C = \sqrt[4]{\frac{E \cdot d^3}{12\,\rho \cdot (1+v^2)} \cdot \omega^2}$$

$$C = \sqrt[4]{\frac{16.10^9 \cdot (0{,}275)^3}{12 \cdot 334\,(1+0{,}3^2)} \cdot (2 \cdot \pi \cdot 100)^2} = 416 \text{ m/s}$$

The dynamic stiffness coefficient

$$fr = 160\sqrt{\frac{s}{m}}$$

$m = \rho \cdot d = 1100 \cdot 0{,}015 = 16{,}5 kg/m2$

$$fr = 160\sqrt{\frac{s}{16{,}5}} \leq 80 Hz$$

$$s \geq 16{,}5\left(\frac{80}{160}\right)^2 = 4{,}13 \text{ MN/m3}$$

The resistance of the wall B

$$R_B = R1 + R2 + 20\log\frac{4\pi f d}{c}$$

$R1 = R_A = 48dB$; $R2 = 20\log \frac{\pi.f.m}{\rho.c} = 20\log\frac{\pi .100.16,5}{1,25 .330} = 22dB$

$R_B = 48 + 22 + 20\log \frac{4\,\pi.100.0,05}{330} = 56$ dB

<u>The Glass area</u>

$Rtotal = R - 10\log [1 + \frac{S\,glass}{S\,total} (10^{\frac{R-R\,glass}{10}} -1)] = 40dB$

$\frac{56-40}{10} = \log [1 + \frac{S\,glass}{S\,total} (10^{\frac{56-35}{10}} -1)]$

$\frac{S\,glass}{S\,total} = \frac{10^{1,6}-1}{10^{2,1}-1} = \frac{38,81}{124,89} = 0,31 = 31\%$

<u>The absorption grad</u>

$R = L1 - L2 + 10\log \frac{S}{A} = 56dB$

$\text{Log}\frac{S}{A} = \frac{56-70+22}{10} = 0,8$

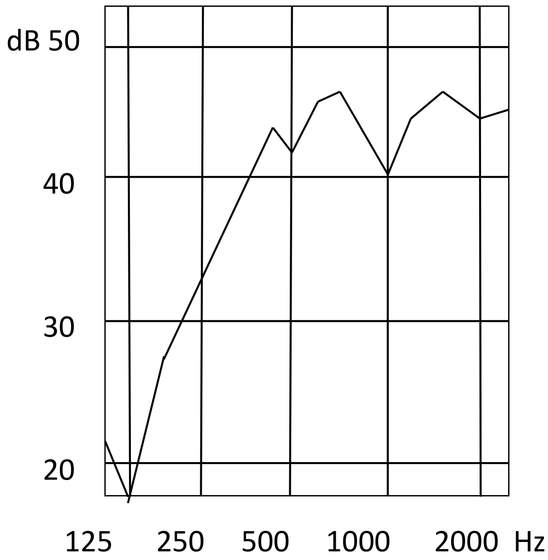
$\frac{S}{A} = 10^{0,8}$

$S = 6,3 . A = 6,3 . \alpha . s$

$\alpha = \frac{1}{6,3} = 0,16 = 16\%$

Problem 2.22

One layering separation wall is between an office and a mitting room. The relation between the resistance and the frequency is like the following diagram.



Given

Mitting room dimension 7 x 4 x 3

Separation wall 4 x 3

Resistance of separation wall 43 dB

Air density 1,25 kg/m3

Air speed 333 m/s

Angle of sound wave 80°

Frequency 1000 Hz

Calculate

- The mass of the separation wall
- The coincidence frequency of the separation wall
- The dynamic stiffness coefficient of the isolation layer to have a resonance frequency of 80 Hz
- The resistance of the wall B
- The absorb area for a reverberation time from 0,5 s

Solution

The mass of the separation wall

R = 37,5 log m – 42 [dB]

$$m = 10^{\frac{R+42}{37,5}} = 10^{\frac{43+42}{37,5}} = 185 \text{ kg/m2}$$

<u>The coincidence frequency of the separation wall</u>

F =125 Hz from the diagram

<u>The dynamic stiffness coefficient of the isolation layer</u>

$$R = L1 - L2 + 10 \log \frac{S}{A}$$

From the diagram

R1000 = 40 dB

S = 3 x 4 = 12m2

40 = 65 – 30 + 10log $\frac{12}{A}$

Otherwise $\frac{12}{A}$ = $10^{0,5}$

A = $\frac{12}{0^{0,5}}$ = 3,8m2

T = 0,163 $\frac{V}{A}$

V = 7 x 4 x 3 = 84 m3

T = 0,163 $\frac{84}{3,8}$ = 3,6 s

<u>The resistance of the wall B</u>

R = 20log $\frac{\pi . f . m}{\rho . c}$. cosv , cosv = cos 80° = 0,17

R = 20 log [$\frac{\pi . 1000 . 185}{1,25 . 333}$] . 0,17 = 47,5dB

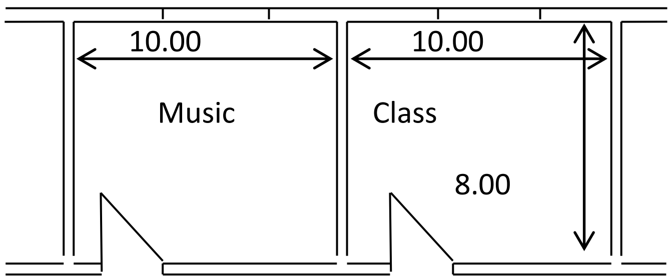
<u>The absorb area for a reverberation time from 0,5 s</u>

ΔL = 10 log $\frac{T1}{T}$ = 10 log $\frac{0,5}{3,6}$= -8,6 dB

A = 0,163 . $\frac{V}{T1}$ = 0,163 . $\frac{84}{0,5}$ = 27,4m2

Problem 2.23

In a school we have a separation wall between music and class room. The separation wall is one layering. The maximum sound intensity by 1000 Hz is 63dB. Please calculate the following:



Given

Height of both rooms 3.0m

Room A reverberation time 1,5s

Room B reverberation time 0,7s

Air density 1,25Kg/m3

Air speed 343 m/s

Frequency 1000 Hz

Angle 45°

Calculate

- The absorb area to have a reverberation time from 0,9s

- The sound intensity, if all music instruments are working
- The isolation of the separation wall between the classroom and the music room, if all music instruments are working and the classroom sound intensity is with 20dB more than before.
- The mass of the separation wall to have a resistance of 50dB

<u>Solution</u>

<u>The absorb area to have reverberation time from 0,9s</u>

$$\Delta A = 0{,}163\ V\ (\frac{1}{T\ after} - \frac{1}{T\ before})$$

V = 10 x 8 x 3 = 240 m3

$\Delta A = 0{,}163 \times 240 \times (\frac{1}{0{,}9} - \frac{1}{1{,}5}) = 17{,}4m2$

<u>The sound intensity, if all music instruments are working</u>

$\Delta L = L_A - L = 10 \log \frac{IA}{Io} - 10\log\frac{I}{Io} = 10 \log \frac{IA}{I}$

IA = 5 I

$\Delta A = 10 \log 5 = 7dB$

<u>The isolation of the separation wall between the classroom and the music room, if all music instruments are working and the classroom sound intensity is with 20dB more than before.</u>

$R = L1 - L2 + 10 \log \frac{S}{A}$

$S = 8 \times 3 = 24m2$

$A = 0{,}163 \frac{V}{A} = 0{,}163 \frac{240}{0{,}7} = 55{,}9m2$

$R \geq (63+7) - 20 + 10 \log \frac{24}{55{,}9} = 46dB$

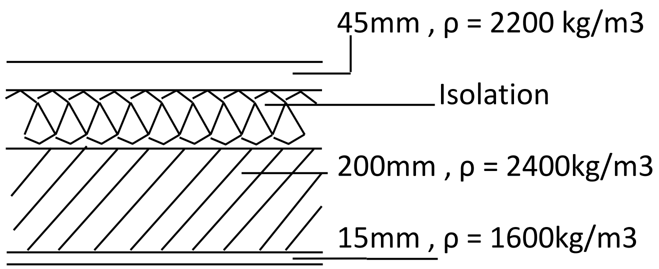
<u>The mass of the separation wall to have a resistance of 50dB</u>

$R = 37{,}5 - \log m - 42$

$m = 10^{\frac{R+42}{37{,}5}} = 10^{\frac{50+42}{37{,}5}} = 284\ kg/m2$

Problem 2.24

A ceiling separate a disco and classroom. Please calculate the following, if you now that, we have 4 boxes in the disco.



<u>Given</u>

Room dimension   10 x 6 x 4

Reverberation time   1,5s

Air density 1,25 kg/m3 , Angle 45°

Air speed 343m/s , frequency 500Hz

<u>Calculate</u>

- The resistance of the ceiling
- The sound intensity if all boxes are working
- The dynamic stiffness of the construction

<u>Solution</u>

$$R = 20\log\frac{\pi . f . m . \cos v}{\rho . c}$$

m =1600 x 0,015 + 2400 x 0,2 = 504kg/m2

$$R = 20\log\frac{\pi . 500 . 504 . \sqrt{2}}{1{,}25 . 343 . 2} = 62{,}3 \text{ dB}$$

<u>The sound intensity if all boxes are working</u>

L1 = 80dB + 10log n

= 80dB + 10log4 = 86dB

$$L2 = L1 - R + 10\log\frac{S}{A}$$

S = 6 x 10 = 60m2

V = 60 x 4 = 240 m3

$A = 0{,}163 \cdot \frac{V}{T} = 0{,}163 \log \frac{240}{1{,}5} = 26{,}1$ dB

$L2 = 86 - 62{,}3 + 10 \log \frac{60}{26{,}1}$ 27,3dB

<u>The dynamic stiffness of the construction</u>

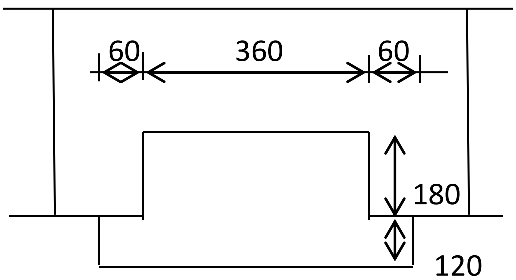
$f_R = 160 \sqrt{\frac{s}{m}} = 85$Hz

m = 2200 x 0,045 = 99 kg/m2

$s = \left(\frac{85}{160}\right)^2 \cdot 99 = 27{,}9$MN/m3

Problem 2.25

The following sketch represent a glass part in a living room. Please calculate the following.



Given

Room height 2,5m

Glass thickness 8mm , Glass density 2500kg/m3

Frame resistance 35dB only 20% from the total area

Reverberation time 1,0s

Air density 1,25kg/m3

Angle in floor 1 45°

Angle in floor 5 75°

Frequency 250Hz , Air speed 343 m/s

Calculate

- The resistance of the glass part in floor1and floor5
- The thickness of the glass in floor 1 and floor 5
- The sound intensity in the glass part if you have in front of your glass part an intensity from 65dB
- The difference ΔL

Solution

The resistance of the glass part in floor1and floor5

$$R= 20\log\frac{\pi . f . m . \cos v}{\rho . c}$$

$M = 2500 \,.\, 0{,}008 = 20 kg/m2$

Without frame

$R = 20\log\frac{\pi \,.250\,.20\,.\sqrt{2}}{1{,}25\,.343\,.2} = 28{,}3$ dB

With frame

$R_{glass} = R_{frame} - 10\log\left[1 + 0{,}8.\left(10^{\frac{R\,frame - Rv}{10}} - 1\right)\right]$

$R_{glass} = 35 - 10\log\left[1 + 0{,}8.\left(10^{\frac{35 - 28{,}3}{10}} - 1\right)\right]$

$R_{glass} = 35 - 10\log\left[1 + 0{,}8.\left(10^{0{,}67} - 1\right)\right] = 29$ dB

If R 45°= R 75°

$20\log\frac{\pi\,.f.\rho.d1\,.cos45}{\rho\,.c} = 20\log\frac{\pi\,.f.\rho.d2\,.cos75}{\rho\,.c}$

d1 . cos45° = d2 cos75°

$d2 = d1\frac{cos45}{cos75} = 8\frac{0{,}707}{0{,}259} = 2{,}19mm$

The sound intensity in the glass part if you have in front of your glass part intensity from 65dB

$L2 = L1 - R + 10\log\frac{S}{A}$

S = 4,8 x 2,5 =12,0m2

V = (4,8x1,2 + 3,6x1,8) .2,5 = 30,6m3

A = 0,163 $\frac{V}{A}$ = 0,163 $\frac{30,6}{1}$ = 39,8dB

ΔL = 10log $\frac{A\ after}{A\ before}$

A after = A before + S ceiling . 0,8 = 5 + (4,8(1,2+ 3,6 .1,8)).0,8 = 14,8m2

Δ L = 10 log $\frac{14,8}{5}$ = 4,7dB

Auditorium Design

Reverberation criteria for speech rooms

The overriding criterion for speech is intelligibility. Since speech consist of short, disconnected sounds 30 to 300ms in length, which are high- frequency, low energy phonemes, the ideal room must ensure the ears undistorted reception of the phonemes. This requires keeping reverberation to a minimum. We can obtain a good approximation of the subjective feeling of livens of a room, for purposes of speech, from the relation

$$T_{R\ (speech)} = 0,3 \log \frac{V}{10}$$

وبهـذه المعادلـة نسـتطيع حسـاب زمـن الصـدى فـي قاعـة محاضـرات أوالعكس بعنى ماهو حجم القاعة المناسب لزمن تـردد كـذا. وقـد يقـول القائل إن حجم القاعة النـاتج هـو بـدون الأخـذ فـي الإعتبـار مـن إضـافة المسطحات الماصة أو عـوازل الصـوت المختلفـة . وأقـول نعـم إن حجـم الفراغ الناتج هو مسطحات معراة من أي إضافة .

For instance, a typical classroom might have a volume of 150m3. Optimum reverberation time is

$$T_R = 0,3 \log 15 = 0,35s$$

Reverberation times longer than this would sound live, shorter ones dead and flat. Indeed, an increase of 20% in reverberation time would make the room excessively live and would negatively affect speech intelligibility.

ويمكننا تحديد زمن الصدى ثم إضافة المعالجـات اللازمـة وحسـاب زمـن الصـدى الجديـد. وفـي جميـع الأحـوال هـذه العمليـة ، أعنـي عمليـة الoptimization تفرز زمـن صـدى صغير وبالتـالي راحـة سـمعية للفـراغ الداخلي .

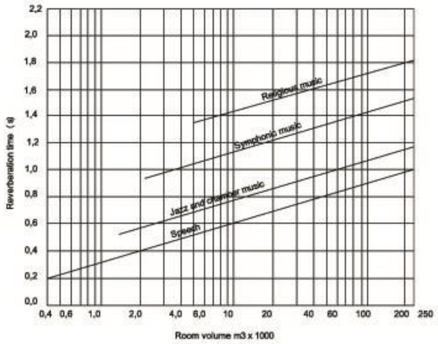


Fig:2.23 optimum mid frequency reverberation times [12]

Figure23 gives optimum mid frequency reverberation times as a function of room size and use figure24 gives maximum reverberation times for speech in large rooms. For good speech intelligibility, reverberation time should remain essentially flat down to 100 Hz

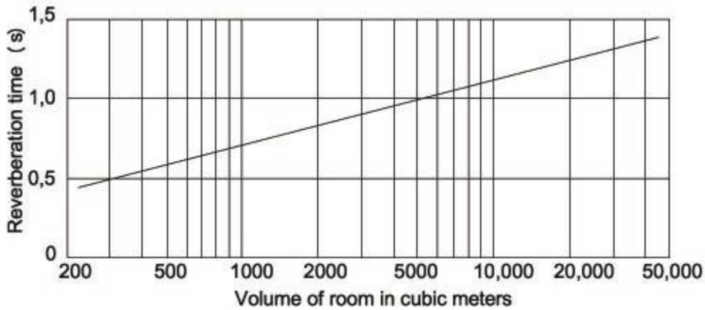


Fig:2.24. maximum reverberation times for speech in large rooms[12]

The reflected sounds associated with reverberation can have either a salutary or a deleterious effect. The ear cannot distinguish between

sounds that arrive within a maximum of 50ms of each other (some authorities use 40ms) .

Sounds arriving within this time reinforce the direct path signal and appear to come from the source. Sounds arriving after this time are apprehended as a fuzzy echo or elongation of the sound, reducing intelligibility and directivity.

<u>وينبغي الأخذ في الإعتبار دوما أن زمن الصدى يزيد بزيادة حجم الفراغ ، راجع الجدول رقم 24، ويمكن تصميم الفراغ صوتيا بالإعتماد على القياسات المترية ، على ألا تزيد المسافة عن17متر بين مصدر الصوت والمتلقي</u>

Since the range of 40 to 50ms corresponds at 344m/s to 13,7 to 17,2m, a speech room should be so arranged that the difference between the first reflection path and the direct path is no greater than 17m and preferably 14m or less at mid frequency (fig25)

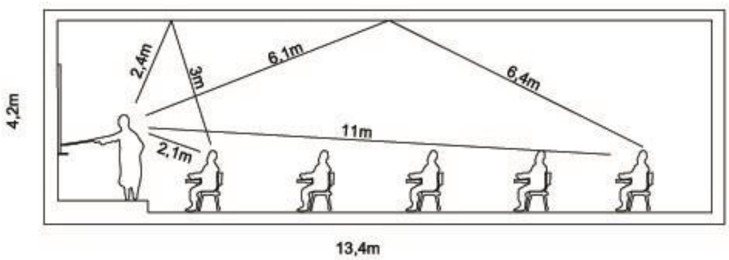


Fig:2.25 sound paths in a typical medium[15]

For more details concerning this and related factors on reflected paths, refer to section sound paths.

Too low reverberation time (very high absorption, minimum reflection) is also undesirable because:

- It limits the size of the room to that which can be covered by direct sound only.
- It disturbing to the speaker, since absence of reflection prevents him or her from gauging the proper voice level and tends to cause excessive effort (shouting, as when outdoors).

Thus, proper design of a room for speech is a compromise between the need for some reflection and the desire to minimize reflection to preserve intelligibility.

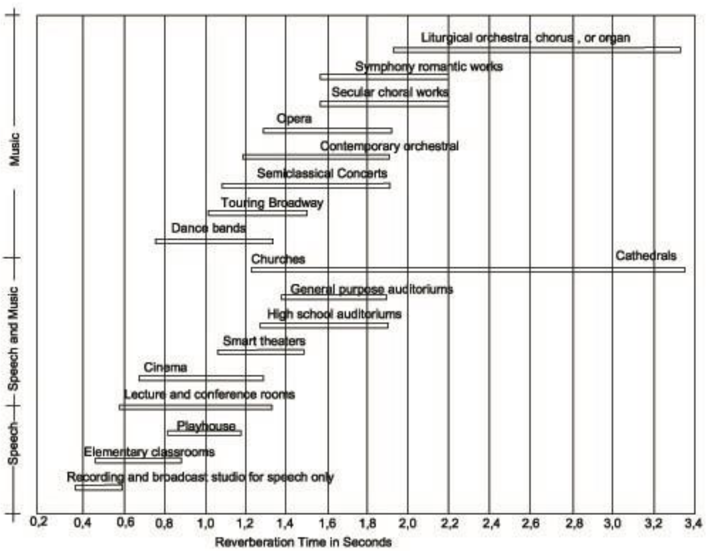


Fig:2:26 Optimum reverberation times a mid-frequencies [18]

Criteria for music performance

Adequate design for a music space requires recognition of the following:

- Large-volume spaces requires direct-path sound reinforcement by reflection.
- Relatively long reverberation time is needed to enhance the music – the exact amount depending upon the type of music. Designers should keep in mind that reverberation time recommendations vary as much as 100% among respected sources.
- It is generally agreed that reverberation time should vary inversely with frequency (T_R should be longer at lower frequencies and shorter at higher frequencies). The longer TR at low frequencies adds fullness to music and body to speech. Thus, TR at 100Hz should be, according to most researchers 35% to 75% longer than TR at the center frequencies
- Short TR at upper frequencies adds directivity gives the sense of depth and instrument location necessary for proper appreciation. This is often referred to as clarity or definition in music. With a solo instrument, this problem is diminished.
- Brilliance of tone is primarily a function of the high-frequency content. Since these frequencies are most readily absorbed, a good direct path must exist between sound source and listener. Since our eyes and ears are close together, a good sound path exists when a good vision path exists. At the other end of the spectrum, lack of sufficient bass expresses itself as a loss of fullness which is often caused by resonant absorption.

The actual design of a music performance space is a very complex procedure involving extensive calculations of absorption,

reverberation time, and ray diagramming, as well as juggling of materials, dimensions, and wall angles. Simulation techniques and acoustic models are also often employed.

Recent research and simulation studies of concert and recital halls have demonstrated that the sensation of fullness of music, or what is today referred to as sound envelopment, is enhanced by lateral reflections that reinforce the direct signal. It has also been found that the subjective judgment of reverberance is more strongly affected by early decay time (the time required for a 10-dB decrease in signal strength) than by conventional 60-dB decay time. Finally, crispness or clarity of the music (particularly important in recital halls and for chamber music) depends upon reflections arriving within 40 to 70 ms. All of these factors are considered both in the original design and in the often length "turning" process of a space intended for music performance. Most modern design solutions also use movable reflector panels and other active variables. After construction is completed, extensive tests are conducted and field adjustments are made.

يلاحظ دوما زيادة زمن الصدى مع نقصان التردد ، والعكس بالعكس . هذا بالإضافة إلى التنبية على أهمية التحكم في أن تكون الموجات الصوتية التي تصل إلى الفراغ الذي نصممه الأفضل أن تكون بين 40-70ms

Sound Path

Ideally, every listener in a lecture hall, theater, or concert hall should hear the speaker or performer with the same degree of loudness and clarity. Since this is obviousty impossible by direct-path sound, the essential design task is to devise methods for reinforcing desirable reflections and minimizing and controlling

undesirable ones. Normally only the first reflection is considered in ray diagramming since it is strongest. Second and subsequent reflections are usually attenuated to the point that they need not be considered except for the special situations of flutter, echoes, and standing waves discussed below

Specular Reflection

Specular reflection occurs when sound reflects off a hard, polished surface. This characteristic can be used to god advantage to create an effective image source. In ancient Greek and Roman theaters, seats were arranged on a steep conical surface around the performers. The virtue of this arrangement (fig.27) is that the sound energy travels to each location with minimal attenuation.

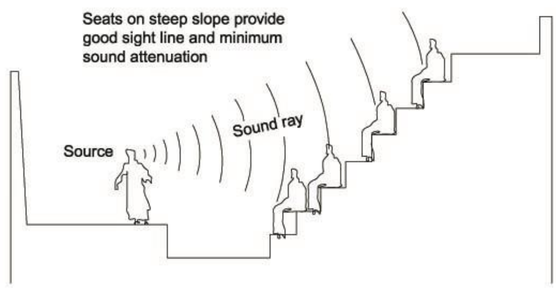


Fig:2.27 creates an image source that stands in approximately the same relation to the audience as the performer in the classic Greek theater. [18]

The same effect can be accomplished by placing the sound source above the seats. This is not practical physically, but it can be accomplished effectively by the use of a reflecting panel (fig.28)

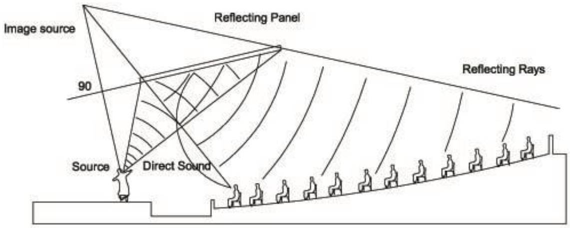


Fig. 28 use of an angled reflector panel [18]

The panel dimension must be at least one wave length at the lowest frequency under consideration. Fig.29 is a chart for converting from frequency to wavelength in feet and meters

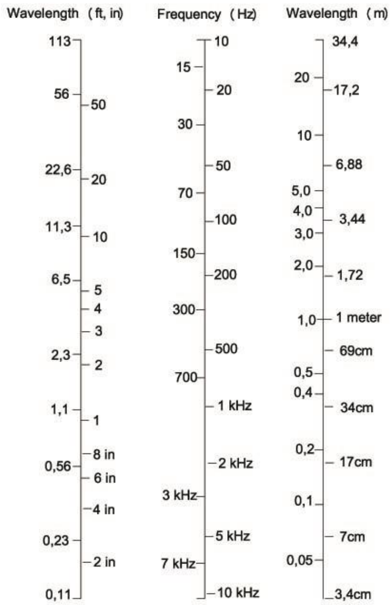


Fig:2.29 monograph for determining wavelength in feet or in meters[17]

لا يمكن بحال من الأحوال الإعتماد على وصول الصوت بشكل مباشر وبنفس القوة داخل الفراغ الواحد، لذا لم يجد الفيزيائيون بدا من الإعتماد على إنعكاس الصوت ، وأصبح تصميم المسطحات العاكسة يترجم المكان الذي نرغب في وصول الموجة الصوتية له .

وليس هذا كلاما جديدا ، فالقباب والمحاريب الموجودة في العمارة الإسلامية تترجم عناصر توزيع الصوت بمعنى

-المحراب يوزع صوت الإمام على المصلين بالتساوي

-القبة توزع الصوت على طلاب العلم الجالسين في الأزهر مثلا بالتساوي

Echoes

As explained in the reverberation criteria, a clear echo is caused when the reflected sound at sufficient intensity reaches a listener more than 50ms after he or she has heard the direct sound. (Some authorities place this figure as high as 80ms). Echoes, even if not distinctly discernible, are undesirable, they make speech less intelligible and make music sound “mushy”. The relative undesirability depends upon the time delay and loudness relative to the direct sound, which, in turn are dependent upon the size, position, shape, and absorption of the reflecting surface.

متى يحدث الصدى؟

يحدث الصدى إذا وصل للسامع صوت غير مباشر بعد 50 ميكروثانية. والواجب على المصمم أن تكون حساباته أقل من هذا الرقم

Typical echo-producing surfaces in an auditorium are the back wall and the ceiling above the proscenium. Fig 30shows theses problems and suggests remedies. Note that the energy that produced the echoes can be redirected to places where it becomes useful reinforcement. if echo control by absorption

alone were used on the ceiling and back wall, that energy would be wasted. The rear wall, since its area cannot be reduced too far, may have to be made more sound absorptive to reduce the loudness of the reflected sound.

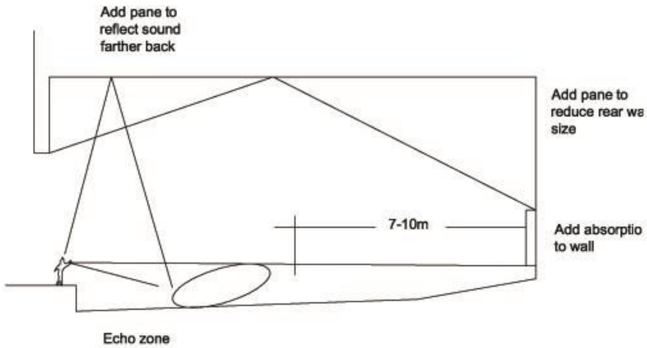


Fig:2.30 auditorium section showing the causes and remedies for two typical echoes [16]

Flutter

A flutter, perceived as a buzzing or clicking sound, comprises repeated echoes traversing back and forth between two non-absorbing parallel (flat or concave) surfaces. Flutters often occur between shallow domes and hard, flat floors. The remedy for a flutter is either to change the shape of the reflectors or their parallel relationship or to add absorption. The solution chosen will depend upon reverberation requirements, cost, and aesthetics.

Diffusion

This is the converse of focusing and occurs primarily when sound is reflected from convex surfaces. A degree of diffusion is also

provided by flat horizontal and inclined reflectors (Fig31). In a diffuse sound field the sound level remains relatively constant throughout the space, an extremely desirable property for musical performances.

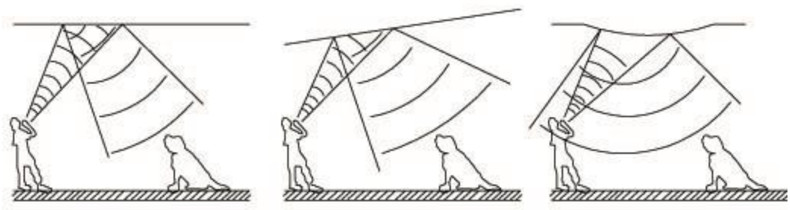


Fig:2.31 sound diffusion can be created with reflectors of different shapes, ranging from horizontal flat, inclined flat or convex [16]

الصوت المنخفض (الوشوشة) وتشتيت الصوت ، الذي الصوت المنخفض أحد عناصرة يحدث باللوائح المنبسطة واللوائح المحدبة ، لكن المحدب منها يوزع الصوت ويشتته بشكل أفضل كما هو موضح أعلاه .

لذا نجد أن اللوائح المحدبة يكثر إنتشارها في المسارح .

<u>Focusing</u>

Concave domes, vaults, or walls will focus reflected sound into certain areas of rooms. This has several disadvantages.

For example, it will deprive some listeners of useful sound reflections and cause hot spots at other audience positions (fig.32)

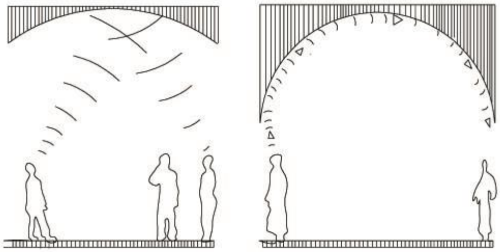


Fig:2.32 Two undesirable phenomena in room acoustics[16]

Creep

This describes the reflection of sound along a curved surface from a source near the surface. Although the sound can be heard at points along the surface, it is inaudible away from the surface. Creep is illustrated in fig.32

Standing Waves

Standing waves and flutter are very similar in principle and cause but are heard quite differently when an impulse (such as a hand clap) is the energy source, a flutter will occur between two highly reflective parallel walls. It is perceived as a slowly decaying buzz. When a steady, pure tone is the source, a standing waves will occur, but only when the parallel walls are spaced apart at some integral multiple of a half-wavelength.

When parallel walls are exactly one-half wavelength apart, the tone will sound very loud near the walls and very quiet halfway between them. This is because at the center, the reflected waves

traveling in one direction are exactly one-half wavelength away from those traveling in the other direction, and are thus equal cancellation. In other rooms, standing waves are noted as points of quiet and loudness in the room.

تتكون الموجات الواقفة بين لـوحين متـوازيين وعاكسـين للصـوت بدرجـة عالية على أن تكون الموجات الواقعة بينهما نصـف موجـة فقـط. وتحـدث هذه الموجات في الغرف ذات الأبعاد الصغيرة.

Standing waves are important only in rooms that are small with respect to the wavelength generated (smallest room dimension < 20 m for music or < 10 m for speech). Another effect of standing waves, called resonance, is the accentuation of a particular frequency that will cause a standing wave. Thus, if one speaks (or plays a music instrument) while standing near a wall of a room about 5mx5m in size , one will notice an abnormal and sometimes unpleasant loudness in the sound at about 280Hz.

Similarly, when a music plays a scale, one note may seem far louder than the adjacent ones, and listeners in one section of the room will hear a quality of sound different from that heard by those in other sections. This effect must be avoided for music performance but is merely an annoyance in rooms designed for speech use. This is one the reasons that one finds music rehearsal rooms, broadcast studios, and the like with nonparallel walls and undulating ceilings. These irregularities prevent most of the undesirable effect described above from occurring.

However, since rooms with nonparallel walls and undulating ceiling are for most applications unacceptable and since standing waves and other resonant phenomena are related to room geometry. It is possible to calculate room proportions with conventional geometries that will minimize these effects.

Fig.33 shows such room proportions, which are applicable to bass frequencies (unit 100Hz)

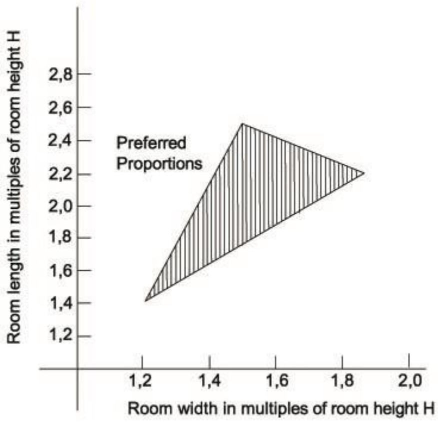


Fig:2.33 triangular figure contains the room proportions recommended, for avoidance of disturbing low frequency acoustic phenomena such as flutter , standing waves and resonances, particularly at frequencies under 100Hz [22]

These are the problematic frequencies, since at higher frequencies (and in large rooms) many standing waves occur, and the total effect is much less disturbing and frequently hardly noticeable.

Ray Diagrams

Ray diagramming is a design procedure for analyzing the reflected sound distribution throughout a hall using the first reflection

only. Fig.34 shows a ray diagram. The rays are drown normal (perpendicular) to the spherically propagating Sound waves. Specular reflection is assumed, that is, at reflecting panels the angles of the incident and reflected rays are always equal. Thus, in addition to direct sound each listener is receiving reflected sound energy. It is as though there were additional sound sources, the real one and numerous image sources.

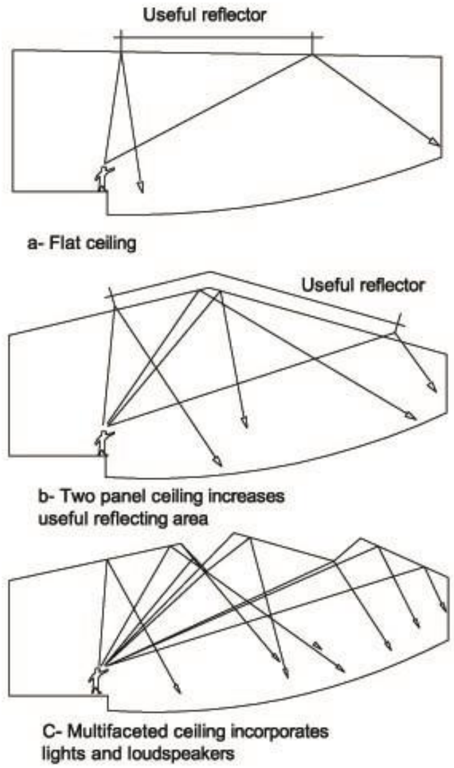


Fig:2.34 section through a typical lecture room showing use of ray diagrams [18]

Fig .34 shows the application of ray diagram to the design of a lecture hall. In fig.34 , the stage height and seating slope are arranged to provide good sight lines, and the ceiling height is established by reverberation requirements, aesthetics, cost, and so on. It can be seen that less than half of the ceiling is providing useful reflection. By dividing the ceiling into two panels (fig.34) , people in the rear of the room perceive the direct source plus two image sources, and the useful reflecting area is increased by 50%. In fig34 , the shape has been further refined to include a lighting slot and a loudspeaker grille. As mentioned previously, the sensation of musical envelopment that is so satisfying to a listener.

Particularly when dealing with large groups playing symphonic music, is due in large measure to reflections received from the side (lateral reflections. This being so ray diagramming, which was originally performed ray diagramming, which originally performed on room sections to determine ceiling shapes and the proper placement of ceiling-hung reflector panels, is now done in plan as well. This is particularly important in fan-shaped auditoriums, where a canted wall dose not provides useful lateral reflections. The solution to this problem is to build a saw tooth wall or to use reflectors panels along the wall that will provide the desired lateral reflection. Nonhorizontal ceiling reflectors panels can also provide a measure of lateral reflectors panels can also provide a measure of lateral reflection. Particularly in balconies. Although they are a useful design tool, ray diagrams have certain restrictions. Design solutions always will require compromise between ray diagram results for various speaking

positions on a stage. Thus, a parabolic may be a perfect shape for one source position but a very poor shape for other positions.

Auditorium Design

Auditorium is a general term used to describe a space where people sit and listen to speech or music. Acoustical design of an auditorium includes room acoustic, noise control, and sound system design. Acoustical environments can be altered by changing the space volume, moving reflecting surfaces, and adding or subtracting sound-absorbing treatment. Fig.35 illustrates several examples of acoustical adjustability.

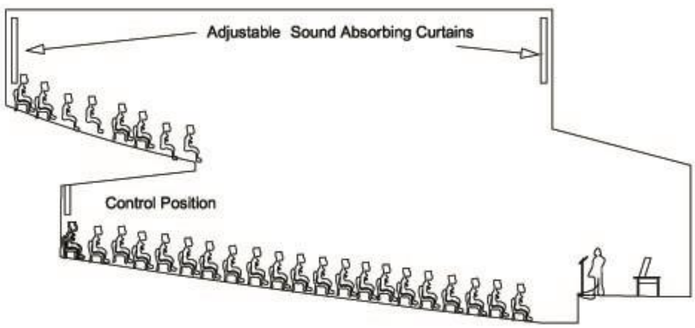


Fig:2.35 adjustable acoustic elements in an auditorium

Factors that influence acoustical design include audience size, range of performance activities, and sophistication of the

potential audience. A small school auditorium and a professional theater will have widely divergent demand from both audiences and performers. The audience size determines the basic floor area of an auditorium, assuming no balconies. Ones this area has been fixed, the volume of the room is developed according to reverberation requirement of the space.

يلاحظ في تصميم المسارح أن لدينا نوعين من المسطحات

- مسطحات عاكسة
- مسطحات ماصة

هذه المسطحات لا توضع بشكل عشوائي ، ولكن لكل منها مكانة كما هو موضح في الرسم أعلاه. وبذا تكون تصميمات المسارح من العمليات المعقدة التي لا يجوز فيها التنازل العلمي.

In fig.36 shows a typical auditorium in plan and section. The wall and ceiling surfaces is developed to provide proper distribution of sound and eliminate focusing of echoes. Essential characteristics of the design include:

- Ceiling and side walls at the front of the auditorium distribute sound to the audience. These surface must be close enough to the performers to minimize time delays between direct sound and reflected sound.
- Ceiling and side walls provide diffusion.

Acoustic must be considered in the selection of materials used in an auditorium. All auditoriums use both sound-reflecting and sound-absorbing material in any auditorium is the audience, the difference in acoustical characteristics that occurs without an audience may minimized by using fully upholstered seating.

Chairs with fully upholstered seats and backs, covered in an open-wave material, will have absorption characteristics approximating those of an audience. Using the auditorium in fig 35 As an example, the reverberation characteristics of an auditorium with various materials may be examined. In the first example, the room use is assumed to be for music performance. The only sound absorption is that provided by the audience and seating. In the second set of calculations, absorptive curtains were installed along the rear wall and a portion of the side wall. This configuration might be used for lectures in a room that is adjustable between speech and music configurations. A third configuration might use permanent sound-absorbing treatment installed on the ceiling and rear and side walls. Because of its low reverberation time, this configuration would be appropriate only for movies and lectures, note for music activities.

Simplified calculation of Mid-frequency (500 and1000Hz) Average Reverberation Times

Reverberation Time (TR) = $\frac{0{,}05\ x\ Volume}{total\ absorption\ area}$

More reverberant conditions (curtains reflected)

	Area	A	absorption
Seating and stage with audience and performers	3323	0,92	3060
Wall area Concrete block	8000	0,2	1600
Lower rear wall Permanent sound absorbing treatment	450	0,88	396
Total absorbing			5056

$$TR = \frac{0{,}05 \times 155{,}500}{5056} = 1{,}5s$$

Less reverberant conditions (curtains exposed)

	Area	A	absorption
Seating and stage with audience and performers	3323	0,92	3060
Wall area Concrete block Balance converged by curtains	3600	0,2	720
Lower rear wall Permanent sound absorbing treatment	450	0,88	396
Curtains	4400	0,45	1970
Total absorbing			6146

$$TR = \frac{0{,}05 \times 155{,}500}{6146} = 1{,}2s$$

These simple examples indicate the effect of changes in the amount of absorption on the characteristics of a room. Adjustable treatments permit the characteristics of the room to be modified to any point between the externs to meet the acoustic program requirements of a multipurpose hall.

وبذا يتضح لنا أن صدى الصوت أوعملية تقليل زمن الصدى لهـا معـاملات أخرى غير حجم الفراغ مثل نوعية تبطـين الكراسـى المتواجـدة ونوعيـة

تغليف الجدران . إن المشاهد البسيط عندما يدخل دارا من دور السينما يجد جدرانها مغلفة بالموكيت. وهذا ليس من فراغ.

Existing spaces may require remedial treatment to eliminate unwanted phenomena such as focusing and echoes, as focusing in fig.36. In the first example, the surface of the dome was converged with sound-absorbing material to eliminate focusing; in the second, sound-absorbing treatment was applied to a curved rear wall to eliminate an echo. Such treatment also will affect the reverberation characteristics

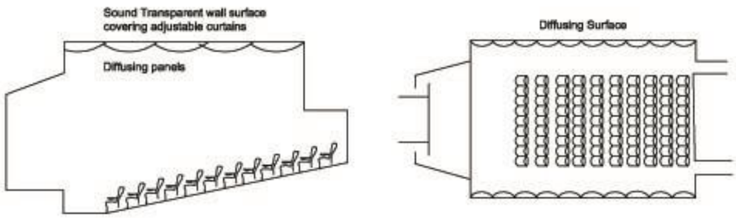


Fig:2.36 simplified calculations of mid-frequency

Sound Reinforcement System

Objectives and Criteria

The purpose of a sound reinforcement system is just what the name indicates to reinforce the sound. This would otherwise be inadequate. Thus, an ideal sound system will give the listener the same loudness quality, directivity, and intelligibility as if the source of sound were immediately adjacent – a distance of 1 to

2m for speech and farther for music, depending upon the type and number of instruments. This situation must obtain for every position in the space within ±3dB. The other factors, loudness and intelligibility have previously been discussed.

By quality we mean that the frequency response should be linear so that reproduced sound bears the same relation between its frequency components as the original sound. (Quality is then field-adjusted by voicing or equalization, as discussed below)

Directivity is the characteristic whereby the sound appears to be coming from the originating source that is the loud-speakers should be directionally “invisible” and the listener must have the impression of actually hearing source. It should be emphasized that sound systems cannot correct a poor acoustic design completely, although they can improve a bad situation.

Generally, sound systems will be required in spaces larger than 1400m3. In terms of occupancy, this volume translates into 550 persons in lecture rooms (5m average ceiling height and 2,1m2 person) and 325 persons in theaters (7m average ceiling height and 2,6m2 per person). In such a room 1400m3 a normal speaking voice can maintain a volume level of only 55 to 60dB, depending upon room design and voice strength.

Components and Specifications

All sound systems consist of three basic element: input devices amplifier, and loudspeaker systems.

Input

Input usually means a microphone, a source of commercial broadcast material of various types, and means for reproducing recorded material in all common commercial formats. Connections to local computers and computer networks are available in sophisticated systems.

Amplifier and controls

Amplifiers must be rated to deliver sufficient power to produce intensity levels of 80dB for speech, 95dB for light music, and 105dB for symphonic music. This assumes a maximum background noise level of 60dB. Thus, 80dB speech intensity will be 20dB higher or four times as loud as the noise level. If the noise level is known to be below 60dB maximum, amplifier and loudspeaker power ratings can be reduced accordingly. The amplifier should carry technical specifications for signal-to-noise ratio, linearity, and distortion. Exact values depend upon the application and are left to the acoustics specialist or sound engineering to supply.

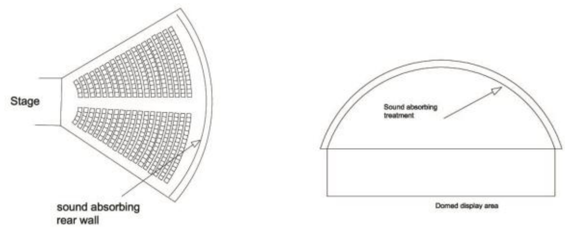


Fig:2.37 Sound absorbing treatment used to eliminate focusing from dome and curved auditorium wall [18]

In addition to the usual volume, tone mixing, and input-output selector controls, the amplifier must contain special equalization controls for signal shaping. These are highly critical filter networks

that, by selective amplification and attenuation of portions of the overall audio frequency spectrum, voice or equalize a system after installation. Equalization is the sine qua non of good sound system; without it, the system will howl, sound rough, give insufficient and poorly distributed gain and sound level, and generally bad sound. Essentially, voicing tailors the system to the acoustic properties of the space. A system not equipped for equalization is not a professional system, and result will verify it. Furthermore, the specification must provide for the services of a competent sound engineer to perform the equalization after installation and construction is complete.

Another control frequency required in theater system is a delay mechanism or circuit that can introduce a time delay into a signal being fed to a loudspeaker. Fig38 show a sound system that covers a majority of an auditorium from a central loudspeaker cluster.

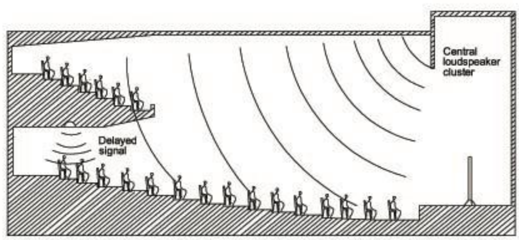


Fig:2.38 loudspeaker system using delayed signal to under-balcony area [18]

The under-balcony seating areas are hidden from the central cluster and receive reinforced sound from distributed

loudspeakers in the under-balcony soffit. To provide directional realism, the signal to the under-balcony loudspeakers must be delayed to allow the weaker signal from the central speakers to arrive first. Delay is necessary because electrical signals travel at the speed of light, whereas sound is much slower (one-millionth of light speed, approximately). With this arrangement, sound will seem to come from the source, and the directivity so necessary to realism is maintained.

Loudspeakers

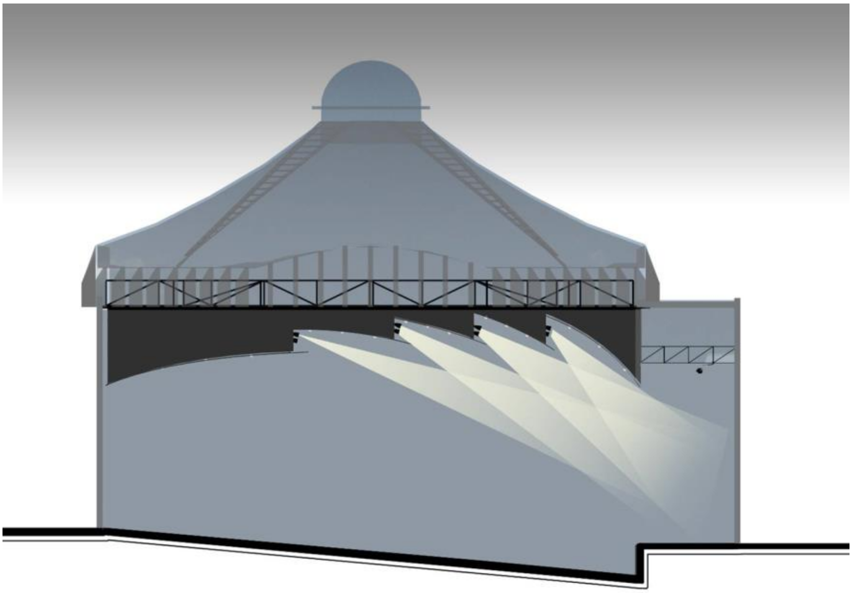
These are the heart of any sound system and obviously must be of the same high quality as the remainder of the system. Indeed, system economics will show up much more quickly in loudspeaker performance than in any other component. Selection of speakers is a complex technical task beyond the scope of our discussion. Nevertheless, a few general remarks are in order. The best system with traditional components uses central-speaker arrays consisting of high-quality, sectional (multicell), directional, high-frequency horns and large-cone woofers. These assemblies are frequently very large, and the architect should be aware of the dimensions that must accommodate. Smaller units with folded horns can be used, at a sacrifice in low frequency response. If only speech is to be reproduced, these units will preform adequately. Distributed system use small (4-12 diameters) low-level speakers. Ceiling-mounted and firing directly down.

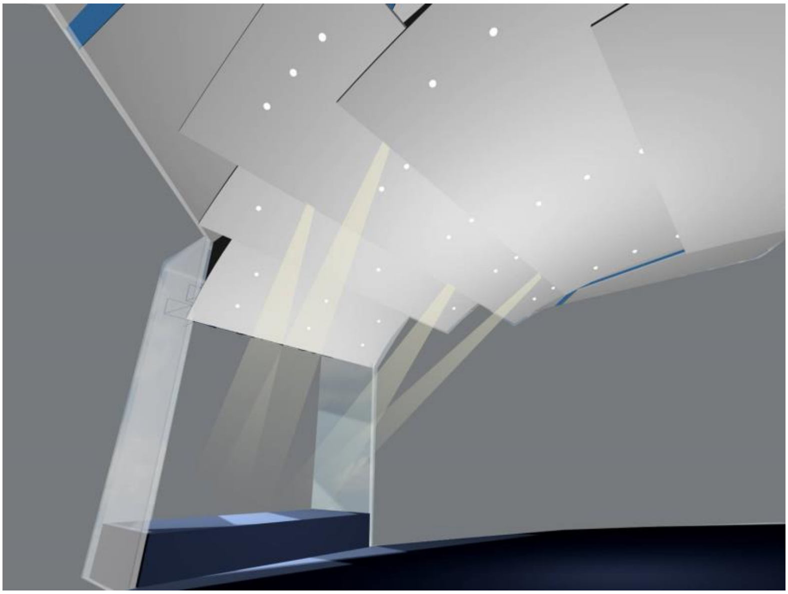
Recent developments in loudspeaker design have produced units much smaller than those previously required for high-power, high-quality, low-frequency sound reproduction. Here again, the considerations are highly technical and speakers should be

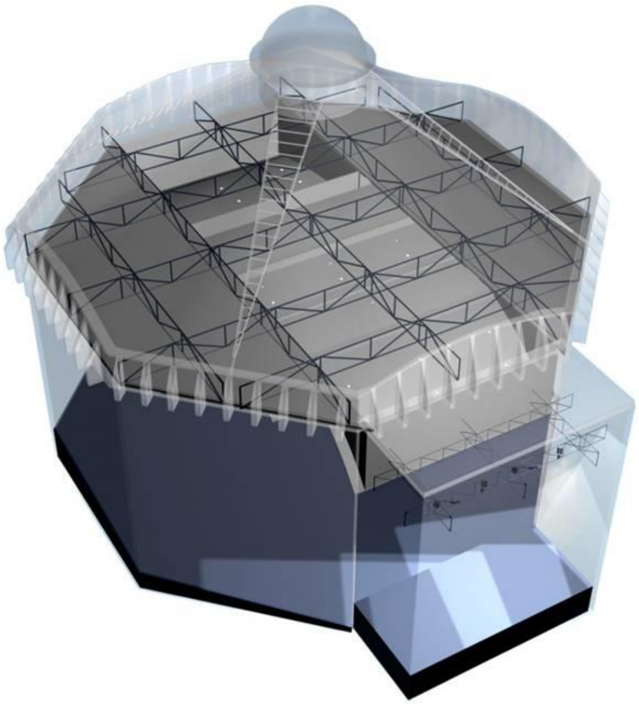
supplied under a performance specification that guarantees user satisfaction subject to specified test procedures.

Selected Dr.Gerisha Projects

MUST Theater



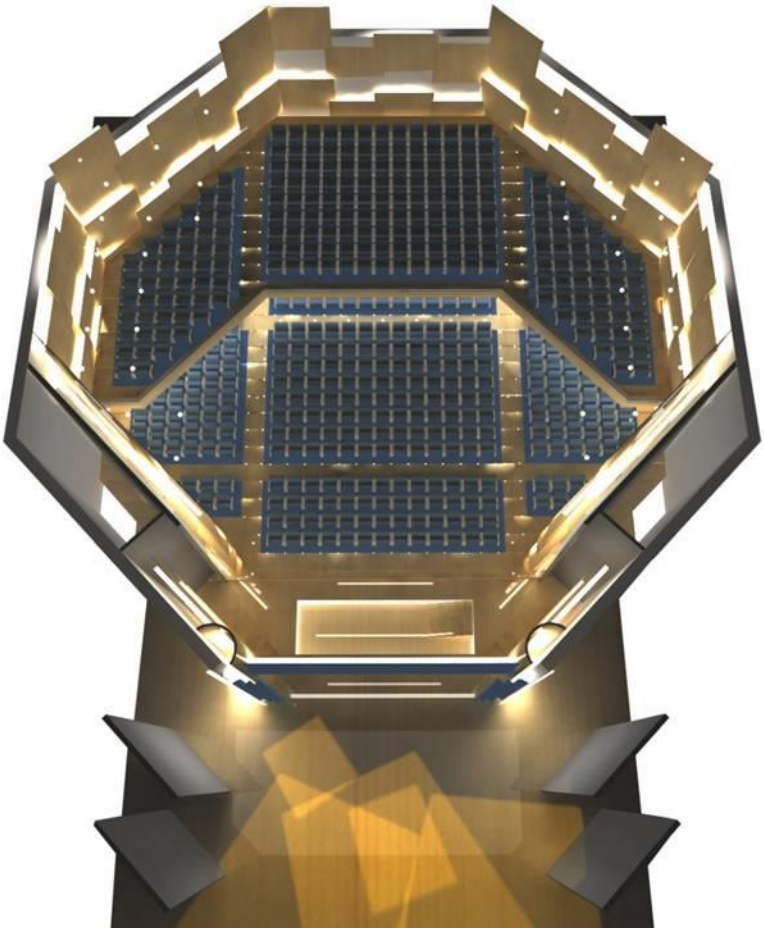




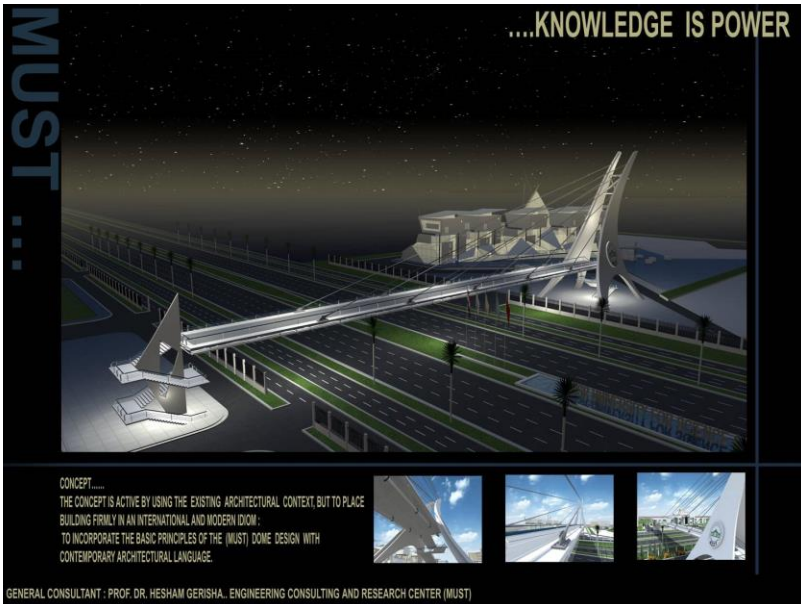
Taken into account in this project all the rules to avoid echo concerning walls and ceiling. We calculated the dimension that should be between the breakers of sound and taken into account the principle of non-parallel walls. The project combines the rules of sound insulation and contemporary interior design.

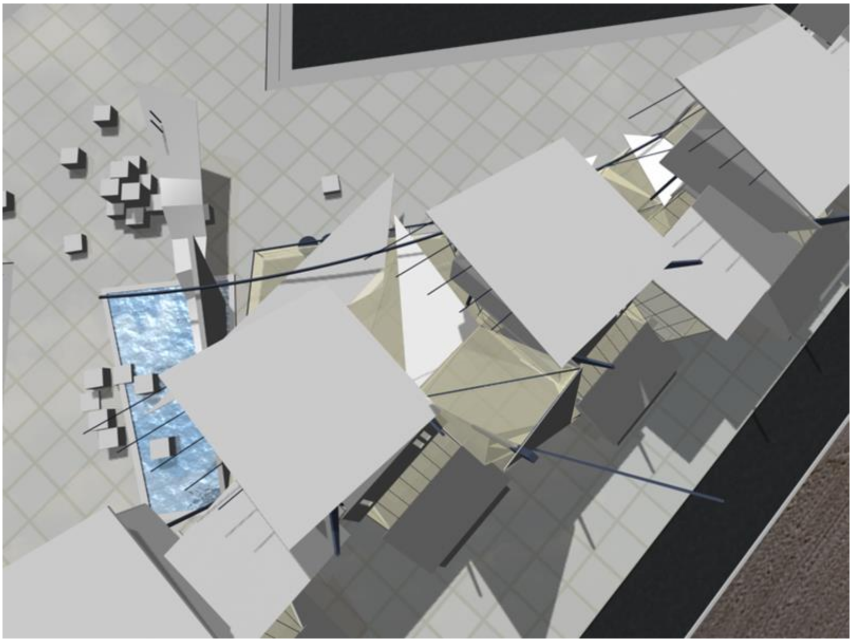
It should be said that we have only the interior designed for this project. Of the difficulties we found soaring levels of the sound ceiling and the top of the dome .

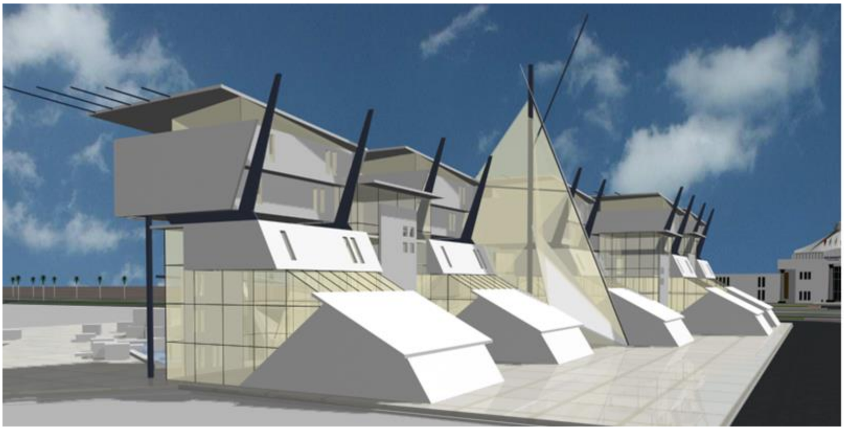

Teachers open the door
but you must enter by yourself...



2.MUST-Extension Project







This is the design for the lecture halls of the faculty of engineering as we sea, the design doesn't include parallel walls to ignore echo.

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Chapter1: Heattransfer